A MANUAL OF

PETROL MOTORS

AND MOTOR CARS.

COMPRISING THE DESIGNING, CONSTRUCTION, AND WORKING OF PETROL MOTORS.

BY

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PREFACE.

In view of the constantly increasing use of motor cars, it is unnecessary to make any apology for the following work, which deals with the design of petrol motors from a practical and commercial as well as a theoretical standpoint. The data on which it is based are the result of long study and practical work, and the principles of construction of the motor car and of modern high-speed internal combustion engines are thoroughly investigated.

The calculations necessary for determining the sizes of the various parts of a car at a given stress, and for calculating the stresses on actual cars and comparing them with those in other machines are also given without unnecessary complications and in such a form as may easily be understood.

The details of construction are also given as fully as possible, both as regards the relative merit and relative cost of different designs. The author would particularly impress on students the necessity for the careful consideration of *all* details of the design, for, however good the general plan may be, a detail not carefully thought out is likely to give trouble and prevent the success of the machine. In the same way, as the cost of a machine is simply the sum of the cost of the parts, if the cost of one of these is unnecessarily high the commercial success of the machine (which should be the primary object) is impaired. It is hoped that the book will be of service to those engaged in motor-construction and to students endeavouring to master the intricacies of motor design, and that it may help to make clearer the technicalities of motor-construction.

While the attitude of the book is necessarily somewhat critical, the author wishes to express his admiration for the various con-

structors of motor cars to whom the extraordinarily rapid development of the industry is due.

He wishes also to express his indebtedness to those makers who have kindly supplied him with particulars of their cars for Tables vii. and viii., and also for the loan of the blocks for Chapter xxi. Also to the various reports of trials, &c., from which the particulars of the other tables are compiled.

F. S.

April, 1907.

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A MANUAL OF

PETROL MOTORS AND MOTOR CARS.

PART I.—ENGINES.

CHAPTER I.

HISTORY AND GENERAL PROGRESS.

WHILE it is not necessary to give the whole history of the petrol-driven car, it is as well to narrate so much of it as will show the steps by which the present general type came into use.

Petrol Car Defined.—A petrol car is a carriage driven by a gas engine using vaporised petrol instead of coal gas, but otherwise exactly similar in principle to the stationary gas engine. As the engine suitable for a car must be light, the number of revolutions per minute must be great, say, from 500 to 2,000. As driving wheels running at this speed would yield an undesirable velocity, this speed must be reduced by means of intermediate gearing. Further, as the road varies a great deal in gradients, &c., either the engine must be so powerful that it can take the car up any hill at full speed or there must be changes of gear. The former plan would entail a very large engine which would only be running at a very small fraction of its power most of the time; the latter, therefore, is the plan usually adopted. In bicycles, however, the engine is usually made powerful, and has only one speed. In cars there is always a change speed of some kind. Various plans have been tried. Several of the early cars had belts driving with a fast-and-loose pulley on to a countershaft which drove the back wheels with chains. This arrangement is shown in fig. 1. The belts were not very satisfactory, as there is not room in the car for them to be made either wide enough or long enough to drive properly. Consequently they were liable to slip. The engine is inconvenient of access, as the cylinder and valves are all under the body of the car. The same defects usually occur in the modifications of it.

In another arrangement, as shown in fig. 2, the engine is at the back of the car with the valves at the after-end. This drives an arrangement of gears by means of a friction clutch. The gears are so arranged that there

are four sets—any one of which can be slid into gear—each giving a different speed. In this case, the valves and ignition are easily reached, but the gearing is very inaccessible. Several types of cars have been made with

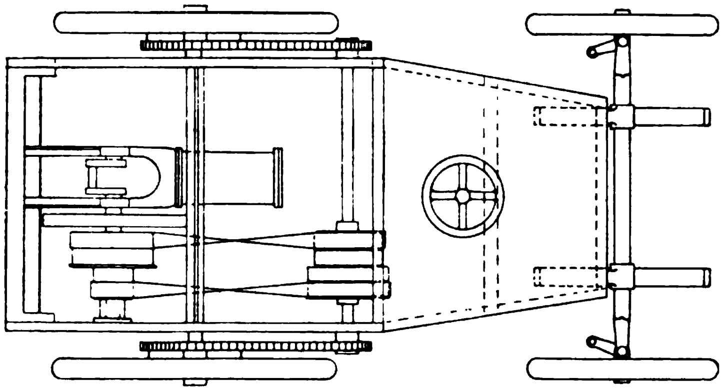


Fig. 1.

a vertical engine under the back seat, either driving the car with face gear, as in fig. 3, or with bevel gear, as in fig. 4. Both these have the engine in a rather inaccessible place, and vibration from the engine is felt much more than when the engine is at the front of the car.

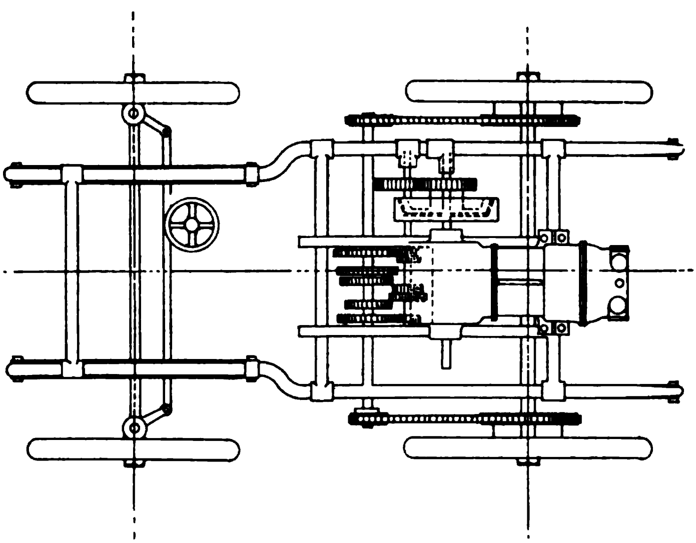


Fig. 2.

All these types are practically obsolete.

The general types of car at present in use are shown in figs. 5 and 6. In these there is a vertical engine in the front of the car, under a removable

bonnet. Behind, there is a gear box containing the change-speed gear. Between the engine and the gear box there is a friction clutch by which they can be disconnected or connected gradually. So far the two types of car are alike. In the first type (fig. 5) there is, behind the change-speed

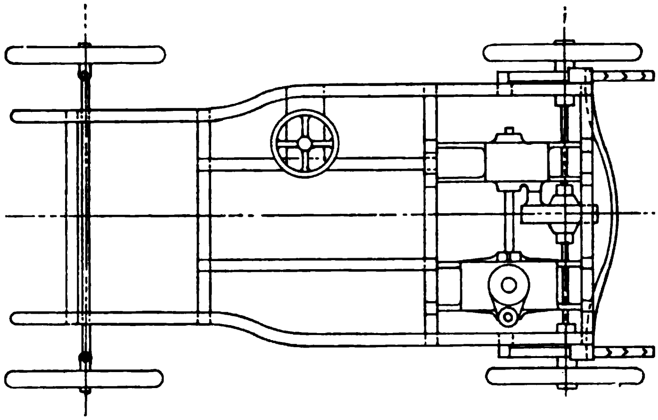


Fig. 3.

gear box, a cross shaft driven by bevel gearing. On this shaft there is the differential gear. From each end of the cross shaft a chain drives one of the back wheels. In the other type (fig. 6) the shaft coming out of the gear box continues in a direct line to the back axle, which is a revolving one, and

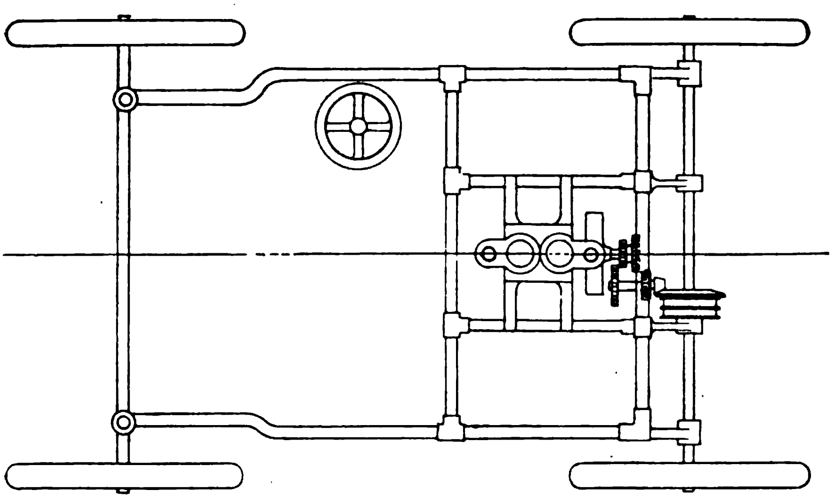


Fig. 4.

has the bevel gear in it to drive it. In the shaft from the back axle to the gear box there are universal joints to allow of the car rising and falling on its springs.

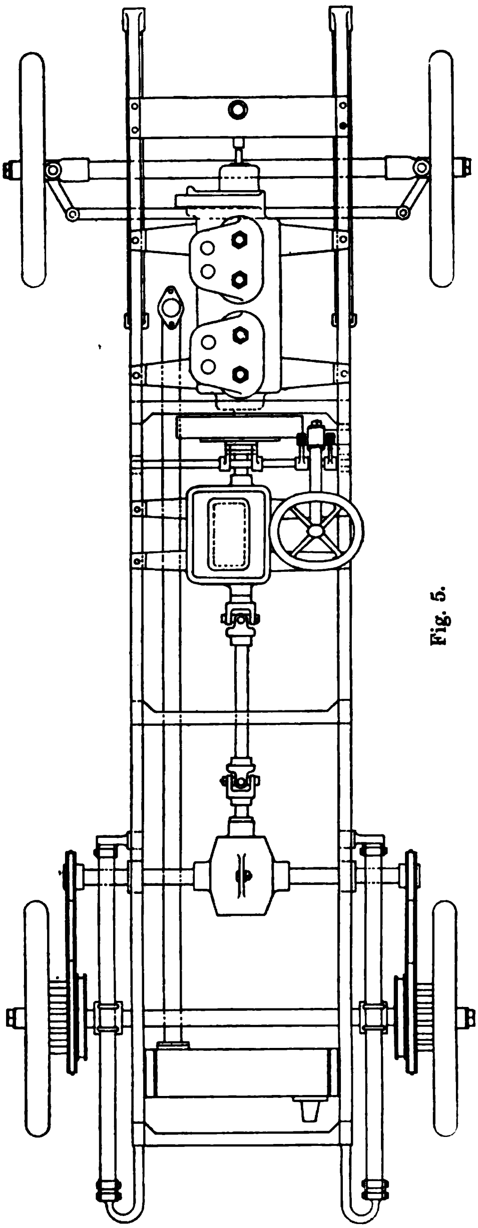


Fig. 5.

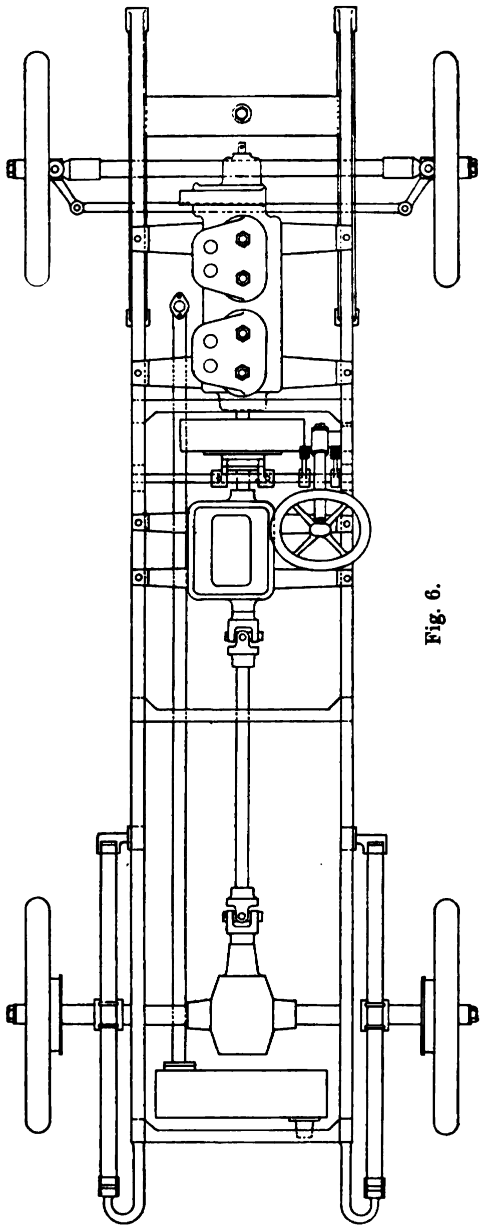


Fig. 6.

Advantages of Ordinary Type of Car.—By far the greater number of cars now made in Europe are of one or other of these types, which are very alike in their main outlines. The great advantages they have over any of the previous types have no doubt brought them into use. In the first place, the engine is obviously very accessible. Should anything go wrong with the ignition, valves, &c., the bonnet can be lifted up or taken off, and everything stands up well from the frame of the car and is easy to get at. The crank case can be reached from underneath; and the friction clutch, as also the gear box, through the floorboards of the front of the car. Further, all the transmission gear is below the floor level of the car, so that the body can be made of any shape independent of the machinery, which is a considerable advantage in manufacture.

Disadvantages of Ordinary Type.—Although these advantages have brought this general type into almost universal use, there are some disadvantages. The fact of having the whole of the engine outside the car adds considerably to the length of the car, and therefore to the wheel base. This again adds to the weight and increases the difficulty of passing round sharp corners.

There are several cars being made at present in which the arrangement is different from these ordinary types. These will be dealt with later.

Will the Present Type be Permanent?—Whether the present type of car will be a permanent one may be doubted. Many of the reasons that have brought it into use may very likely cease to be of any great importance as cars get more used and their various parts more reliable. Further, the demand for racing cars has undoubtedly had somewhat to do with the almost universal adoption of the ordinary type. In a racing car, however, there is practically no body to provide for, and there is no objection to the engine and gear box taking up practically the whole of the car. In a racing car, in fact, the engine is not really in front, but nearly in the middle of the car.

On the other hand, every increase in power in a touring car makes the disadvantages of the forward position of the engine more evident. As long as the engine was a comparatively small one it did not take up a great proportion of the length of the car. Now, however, the engine takes up a considerable amount of room, and the result is that the wheel base is becoming of inconvenient length.

If the engine is put under the car the latter is much better balanced, as the seats are more forward, and therefore all the weights are near the middle of the car, while the balance is much less disturbed by the load; whereas, in the ordinary type of car, the load comes almost entirely on the back wheels, and therefore the proportion of load carried by them is much increased. With the engine in the usual position, the whole of its weight comes on the front wheels, and consequently the body has to be placed very far back to get enough weight on the back wheels, and this makes the seats much less comfortable than when well between the wheels. Further, having a heavy engine at one end of the car and heavy load at the other to balance it, probably makes the car much more liable to skid than when the weights are all concentrated in the middle of the car.

Again, the advantages in the engine being very accessible will, to a great extent, cease as the engines get more reliable. We can see a somewhat analogous progress in the early locomotives. For a long time the standard type in use had a vertical engine placed on the back of the boiler, so that all

the gear was absolutely accessible, and probably, with the crude details of the early designs, this was the best place. On the other hand, when the engines got more perfect, they were placed under the barrel of the boiler, where they are certainly not so convenient to get at, but in every other way are better placed.

This matter will be dealt with more completely in a later chapter on Special Types of Cars.

Development of Heavy Type of Car.—The history of the ordinary type of car for the last ten years has been very largely one of increase of power in proportion to the weight. The early Daimler and Panhard cars had two cylinders about $3\frac{1}{2}$ by 5 inches, and the cars weighed something like 25 cwts. The engines ran at about 700 revolutions per minute and had cut-out governors, to keep them constant at this speed, and tube ignition. The moving parts were not very light, and the engines were not suited for running very much faster than this speed. The early ones gave 4 horse-power, but, by improvements, this was soon increased to $5\frac{1}{2}$. The car had four speeds, with a shaft-to-shaft drive and run-through gear, and the gear wheels were very narrow, the top speed wheel being about $\frac{5}{8}$ inch wide.

Motor Tricycles.—On the other hand, De Dion started building motor tricycles, and, after experimenting with a steam one, he took up the petrol engine. In order to reduce his tricycle to a reasonable weight, he ran his engines very fast; in fact, as fast as they could be made to go. In order to make them satisfactory at this very high speed he made his moving parts as light as possible, and, as a result, he got most remarkable powers out of very small engines. One of his early engines had a cylinder about $2\frac{1}{2}$ by $2\frac{3}{4}$ inches, and gave $1\frac{3}{4}$ horse-power. This made a very successful tricycle, and, in fact, it was when the powers of tricycles were kept moderate that they were most popular. The horse-power soon rose to $2\frac{1}{4}$, then to $2\frac{3}{4}$, $3\frac{1}{2}$, &c., until some racing tricycles were made of 14 horse-power, or more than double that of the early cars of a ton weight.

The tricycles had engines fitted with electric ignition with a single-contact coil and no trembler, and were controlled by a throttle valve worked by hand. Generally speaking, their stroke was equal to their diameter or nearly so.

Development of Light Cars.—Early in the day makers, such as Renault and the Argyll Company, started making cars on very different lines to the large ones, of which the Panhard and Daimler were the type. They used the tricycle type of engine, which was very soon made water-cooled. That is to say, they used the high-speed engine with electric ignition of the single-contact type and throttle control. They also adopted three speeds, with a direct drive on the top speed, the latter not being too high, so that a considerable hill could be taken on the top gear. In the very early heavy cars the top speed could only be used on an absolutely flat road in good condition. The light cars were fitted with pneumatic tyres as a matter of course, and in most cases had a live axle, whereas the heavy cars had side chains. Other makers, such as Aster, &c., began to make engines of the high-speed type, and these were very soon increased in size considerably. Meanwhile the makers of large cars had introduced four-cylindered cars, with cylinders the same size as in the two-cylinder cars, but using most of the same parts, such as axles, &c., thereby considerably increasing the proportion of power to weight. They also took up electric

ignition, though with the ordinary low-speed trembling coil, and hence could not get good power out of their engines when running at a high number of revolutions.

Since that time the two classes have been approaching each other, till there is now little difference between the two. The makers of the heavy cars have put light ones on the market, and the makers of light cars have put larger ones on with two and four cylinders, while each has adopted the improvements introduced by the other, till now there is practically no essential difference between them. One of the most interesting of the earlier class of cars was the Mors dog-cart, which included a four-cylinder engine with electric ignition, and controlled entirely by a throttle valve, the engine running over a great range of speed. The ignition was low-tension, and, therefore, by a single spark; in fact, the car included many of the features of the modern car. The ignition was worked with an accumulator to start the engine and a dynamo for running; the details were not entirely satisfactory, but the principle was the same as is used with great success to-day. The use of the magneto in connection with the low-tension ignition followed largely on the success of the Mercedes cars, and has come very largely into use; while those who fit the high-tension form have also found it desirable to fit them with magneto in place of battery. The throttle control has come very largely into use, being now usually worked by a governor; and electric ignition has become pretty well universal. Although the leading makers of large cars have not taken up the single-contact coil, which was introduced by the tricycle engine-makers in connection with batteries or accumulators, they have used high-speed tremblers on their coils, which have much the same effect. Further, they are now largely using magnetos, either direct with low-tension ignition or in connection with an induction coil, to give high-tension current, in both cases generally working with a single spark.

Present-day Type of Cars.—In this book, therefore, the cars considered are such as are now in general use, in which the engine is vertical in front, the transmission by friction clutch and change-speed gear is of the sliding type, and the drive is either by chains or universal jointed shaft to the back axle. This is done, not because it is assumed that these features will be permanent in "the car of the future," but because they are general in the car of the present. The intention of the book is to describe generally every constructive part of the car, but no endeavour will be made to notice all the possible variations of the general plan. A brief reference to some of the variations will, however, be reserved for a special chapter.

Room for Improvement.—That there is still ample room for improvement, both in the reliability and durability of cars, is obvious. Machinery of various kinds will, as a rule, run continuously for six days a week and nine hours a day, and be serviceable for many years. For instance, the engines of steamers often run at full power for three or even six weeks without stopping, and factory engines will do so for over six months. Many locomotives have worked for more than twenty years with unimpaired efficiency. Motor cars will not last anything like so long, although it should be possible to make them quite as durable.

All new industries go through three stages. In the first or experimental stage the machines are only usable by experts. In the second they are usable by those having an ordinary knowledge of mechanics, but are not perfect machines; and in the third stage the machine is adapted for general

use. Petrol cars have now advanced far in the second stage. There are plenty of good serviceable cars on the market, and the question is which is the cheapest and most reliable. When only a few firms could make a car that would work at all, prices and profits were high, but this condition of things has long since passed away.

A study of some of the leading designs will show that makers are now endeavouring to reduce the cost without diminishing the efficiency of the cars. The competition now in progress will compel the makers to study economy as regards every part of the car consistent with good work, but the order of consideration should be (1) efficiency and durability; and (2) economy of production. This latter point applies more especially to the general designing of a car. For instance, it is very important to avoid an unnecessary number of pieces and needless accuracy of adjustment.

In particular, designs should be well considered in connection with the economy in erection. It is usually much more expensive to do work in the erecting shop than it is on the bench. Therefore, cars should be so designed that there is as little erecting as possible. A study of cars at any exhibition will show anyone acquainted with shop management that some cars have parts so arranged that there are very few pieces to erect on the frame, various small parts, such as pedals, &c., being carried on the larger parts, such as gear boxes, &c., and that in others there are many small pieces which have to be bolted on to the frame. The latter often have also to be carefully set in line with something else, and this is all costly. Again, it is much cheaper to erect parts if they have not to be mathematically in line. Thus, if we have a flexible coupling between the engine and gear box, we need not line it up with that absolute accuracy that is necessary if we do not, and this saves much time. On the other hand, it is often possible to arrange that parts which have to be in line are attached by machined facings, in which case they do not require to be lined up at all.

Into the matter of equipment and management of the factory it is not the province of this book to enter, as it deals with designs only. A well-equipped shop is, however, naturally as necessary for the production of motors and cars as anything else.

Standardisation.—It is admitted generally that it is desirable to "standardise" cars and other machinery which is manufactured in large quantities as far as possible. By some, in fact, this is a sort of stock phrase that is supposed to cover a multitude of sins. In getting out designs, however, the matter has to be very carefully considered, or it is not practicable to carry it out. In putting motors and cars on the market they should be designed on some well-arranged plan with a view of the greatest possible variations of choice to the customer with the least possible number of patterns for the shop to work to. For instance, if it is intended to make engines with varying numbers of cylinders, all the parts should be so arranged as to be, as far as possible, the same for a given sized cylinder whether in an engine with one, two, or more cylinders. There is, in fact, no reason why one cylinder, piston, &c., should not be a standard for that size for all numbers of cylinders. Many details, such as ignition gear, &c., may be the same for all sizes of engines within reasonable limits. This will also reduce the number of spare parts to be kept in stock and the cost of making them, as they can be made in larger numbers. This needs care, however, and if it is intended to do it, all the designs must be roughed out at first, or we shall find that what is suitable for one number of cylinders will not make

a nice design for some other, and a special one has to be made. Wherever the two parts of gear, such as the two halves of cross shafts, axles, &c., can be made the same, instead of having the right- and left-hand parts different, patterns are reduced and the manufacture cheapened.

Again, it generally conduces to economy (1) to avoid multiplying the number of designs; and (2) to make the parts suitable for as many designs as possible by a judicious system of standard sizes. Thus, several axles and engines could be assembled on different frames as required. The axle of a car ought to be made proportionate to the weight on it, and not to the engine power that drives the car. If the width of frames, centres of springs, and wheel gauge were made the same for all types of car, the axles and engines could be assembled in any way preferred without special patterns. At present the majority of makers have a standard axle and frame for a given sized engine, and, therefore, if a very light car with a powerful engine, or *vice versa*, is wanted, it has to be specially made at great expense. If, however, all were standardised, as suggested, any engine and any gear box could be put on any frame with any axles without special patterns.

Specialisation.—There are two kinds of specialisation. In the one only a few patterns of cars are made of the same general design; in the other the car is constructed of parts made by different makers who are specialists for those parts. Both have their advantages and disadvantages and neither is ever carried out absolutely. No maker, for instance, makes all the special parts, such as coils, tyres, electric wires, or makes his own steel, &c.; while, on the other hand, no one would make an engine by buying pistons, cylinders, &c., from different firms. The general and the best practice is to make the car and engine in the same factory; but in some cases this is not so, the engine being made by a maker of engines only, but most of those who started by making engines only are now supplying complete chassis. This is exactly what has taken place in other trades, as, for instance, the shipping trade, in which at one time ships and their engines were generally made by different people, but are now almost always made by the same, the shipbuilders either having put up engine works of their own or amalgamated with engine builders. The reason is obvious on consideration. It is far more important that any machine should be of one uniform design than that each part in itself should be good. That is to say, however good individual parts may be, the machine will never work well if they do not suit each other and make a harmonious whole. I believe, therefore, that in the future practically all the cars will be made by makers who make the whole of the cars themselves. How far they will actually produce the parts in their own factory is purely a question of factory management, and in many cases stampings, castings, &c., will be made by those who do this sort of work specially, but to the design of the car-maker, not their own. I fancy the tendency will be for the larger makers to do the greater part of the work themselves, putting up plant for the purpose as is done in other trades. Whether car bodies should be made by the car-maker is a rather more questionable matter, but generally it is found much cheaper to have a standard body made for a particular chassis.

On the other hand, there are at present certain advantages in small makers being able to buy axles, engines, &c., from which they can make special cars for customers to suit special requirements. Further, there

may for some time be a good trade done in complete sets of parts being supplied to coachbuilders, &c., who will make the frame and body and put them together. This is rather analagous to the bicycle trade, in which a very large trade is done in this way, the wholesale manufacturer doing the machine work in large quantities to gauge, while the retailer fits the parts together and carries out special ideas for his customers.

Few Patterns Needed.—Firms that make complete cars must, in order to ensure success, confine themselves to a few patterns, and pay special attention to making improvements of these; if different classes of work are undertaken there should be a separate factory for each class. The most successful firms up to date seem to have been those who, when they started on a particular type of car, stuck to making that and nothing else until they had made it sufficiently perfect to have a big trade in it.

Lightness and Durability.—In the matter of lightness and durability we are between two stools. To make a car with all the parts so made as to last many years without repairs or renewals would be quite possible, but its weight would be probably prohibitive. The wear on tyres is often the most important part of the upkeep of a car and this largely depends on weight. Therefore, it may often be worth while to accept the necessity for frequently replacing or adjusting working parts in order to keep the car light and reduce the wear on the tyres. The whole object of design should, however, be to make the important wearing parts as strong and substantial and the car as light as possible. Thus, the actual weight of bearings, gear wheels and shafts is generally a very small proportion of the total weight of the car, which is largely made up of the cases containing these. The tendency of late years has been to make the essential parts much more substantial, to reduce the weight of the casings, &c., and to reduce the total weight of the car. This can, doubtless, be continued, and should be considered very carefully.

One point in reducing both cost and weight is to keep the number of parts as small as possible. It very often happens that a machine which looks comparatively heavy is really lighter than one that looks light, from the fact that it has few parts. The machine with a large number of light parts looks very light, but often is really heavier than one with a small number of heavier parts. The latter is also, generally, the most durable and reliable, and is almost always the cheapest to make.

Another point is that very often weight can be saved by making parts compact. This is especially the case with engines. By reducing the length of an engine the weight of the crank and crank case can be lessened, as also the lengths of the bonnet and the frame. If there is a casing under the engine the weight of this is also reduced. As a long frame has to be made of stronger section than a short one to carry the same weight, the weight of the frame is lessened by a good deal more proportionally than the length is. This is a matter that requires a good deal of consideration, as it does not follow that because one engine is lighter than another of the same power it will make a lighter car. If it is longer than the other it may, on the contrary, add so much to the weight of the bonnet, frame, &c., that the car itself is heavier. For this reason some methods of construction which add to the length of the engine, such as making

the cylinders of steel with copper jackets, do not diminish the weight so much as expected.

Naturally, in order to get the best results for the weight the arrangements in the bonnet should be as compact as possible. The practice of some makers of small cars of putting a very small engine in an enormous bonnet unnecessarily adds both to the expense and to the weight. In the larger sizes also the bonnet should be as short as possible in proportion to the length of the engine, but this must not be carried so far as to render the engine inaccessible. The engine should always be easy to get at from the side.

CHAPTER II.

POWER REQUIRED.

Tractive Effort required.—In engineering work the usual plan is to ascertain what power is required for driving the machinery and to construct an engine of this strength; but this is seldom done in the case of cars.

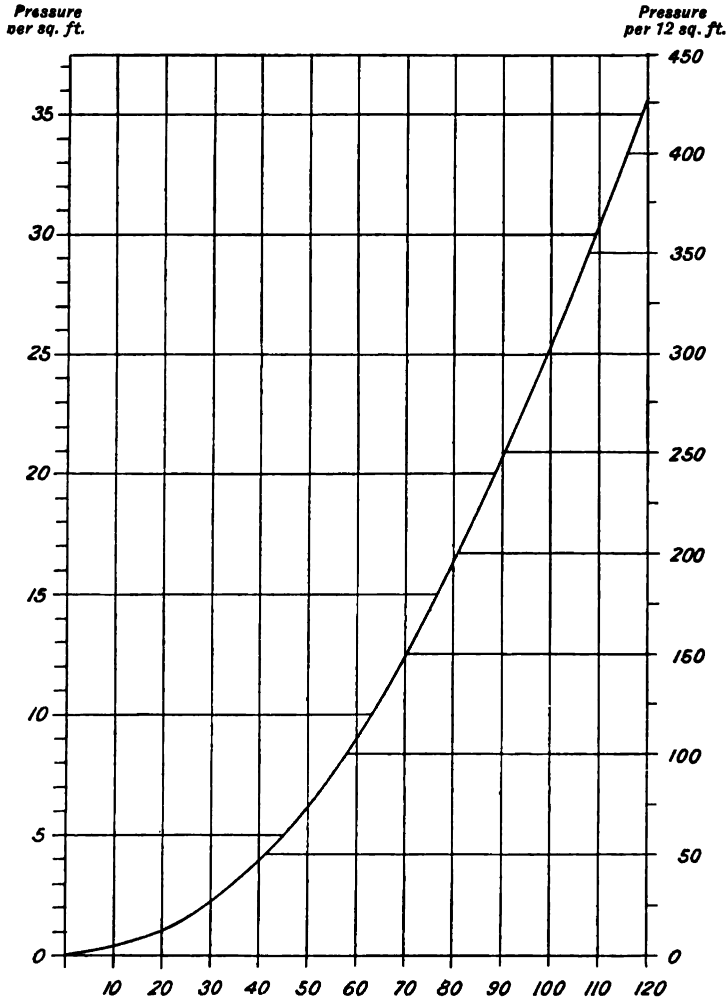


Fig 7.

Frequently the car is made with an engine of some size that suggests itself to the designer, fitted with the gearing seen on some other car and then put on the road. If it does not perform properly the design is altered, at great expense. It would be far better and less expensive if working plans of the proposed kind of engine were first made on paper, based on careful calculations of the power and gears required.

The first point to settle is the force required to move the car. Road surfaces vary so much that it is not possible to give any hard and fast rules; the resistances varying from about 8 lbs. a ton on a railway to over 200 lbs. a ton on a very bad road. On an ordinary road it is between 40 and 100 lbs. a ton, according to whether the surface is dry or muddy, &c. In the

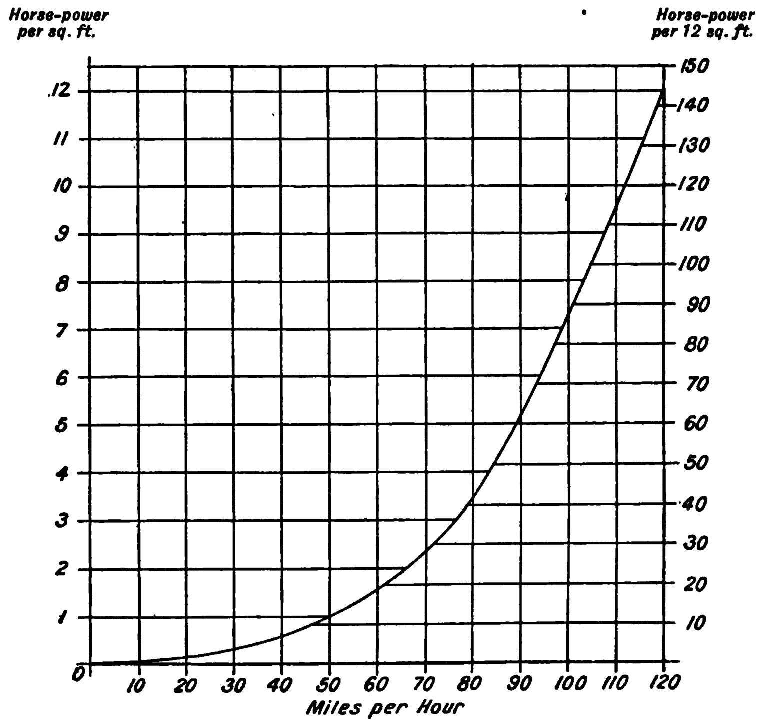


Fig. 8.

small car trials in 1904, the Committee of the Automobile Club estimated the road resistance at 60 lbs. a ton.

Probably a road resistance of 50 lbs. a ton for a pretty good road may be taken as a safe basis for a calculation. The only other important factor in resistance on the flat is wind resistance. In this case the formulæ give different results. Moreover, it is not an easy matter to estimate what the effective area of a car is, as very little of the front of it is flat, and the formulæ are mostly given for perfectly flat surfaces.

According to some data kindly given me by Colonel Crompton, the wind resistance is indicated by the curve shown in fig. 7. This curve represents

the resistance in lbs. per square foot, as also the total resistance for an area of 12 square feet, which is about that of the exposed part of an ordinary uncovered car. Other formulæ, as a rule, give higher rates of resistance than this, but the data on which this is founded was obtained by towing a

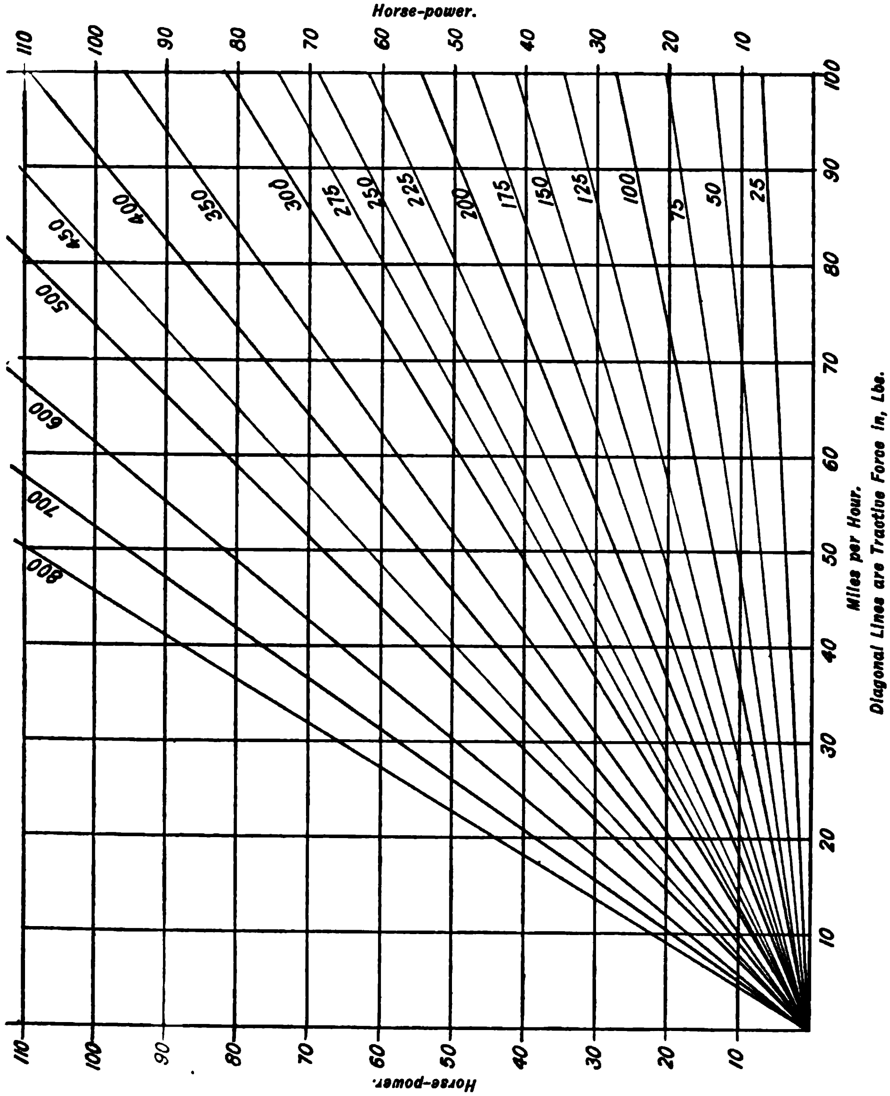


Fig. 9.

vehicle on the road, whereas in most other cases the formulæ were obtained from observations giving the pressure of wind on flat surfaces. Judging from the results actually obtained by racing cars, the curve given seems to correspond pretty well with the actual facts. In order to convert this

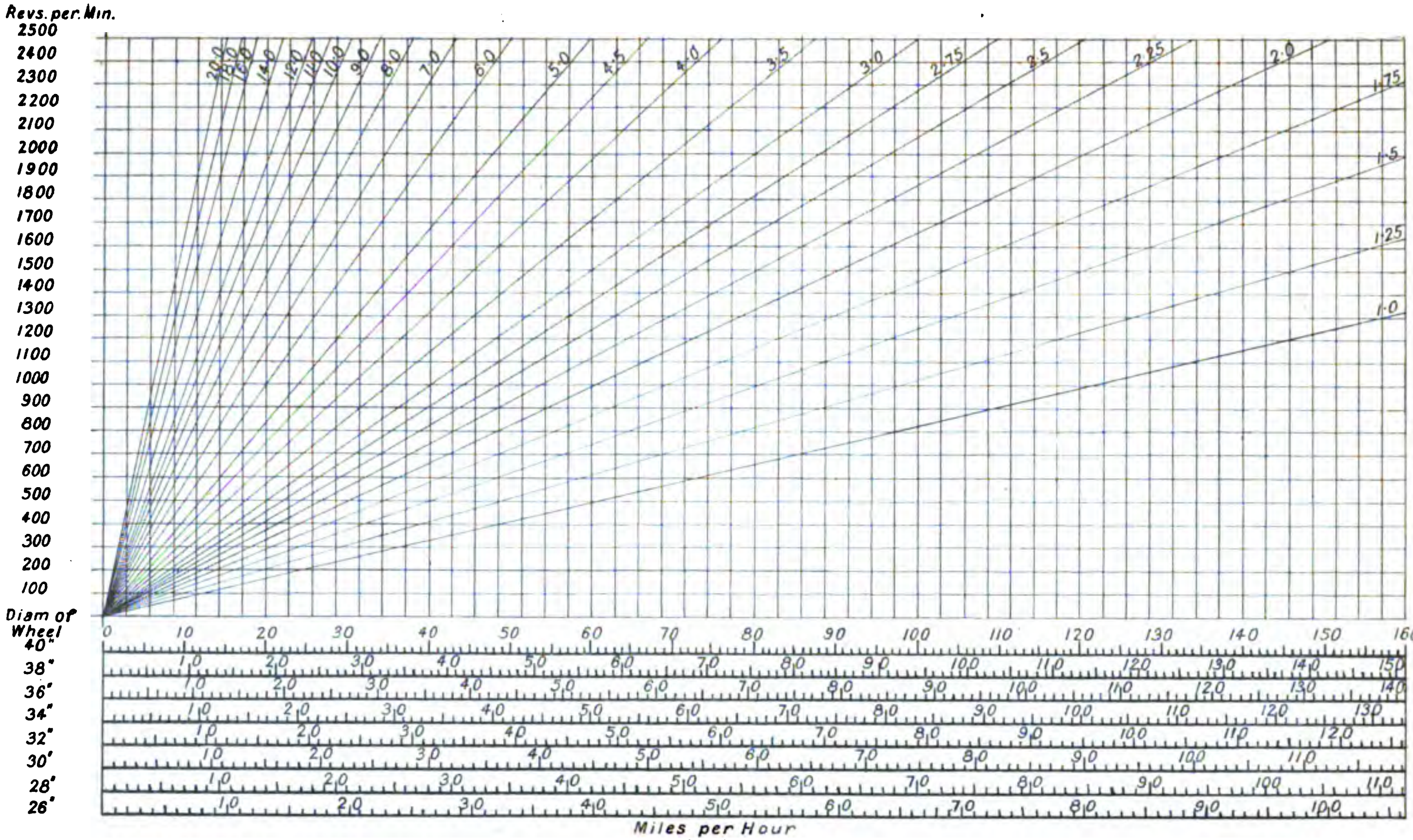


Fig. 10.

The diagonal lines represent the ratio of gear between the engine and the road wheel.

To use Table :—Find the point where the gear ratio and engine revolutions cross and carry this point vertically down the scale opposite road wheel diameter.

Example.—Engine, 1,200 revs.; gear, 3 to 1; road wheels, 36 inches diameter = 42½ M.P.H.

resistance into horse-power multiply the resistance by the speed, and this gives the wind resistance curve shown in fig. 8. It will be seen that while the resistance increases as the square of the speed, the horse-power necessarily increases as the cube.

To calculate the power needed for a given speed with a given tractive force, per ton, the last should be multiplied by the speed and the weight, the formulæ being :—

$$\frac{\text{Tractive force in lbs. per ton} \times \text{speed in feet per second} \times \text{weight in tons}}{550}.$$

Fig. 9 shows the power per ton for different tractive resistances, and speeds.

Example.—What power is required to propel a car weighing 25 cwts. at 35 miles an hour on a good road? Let 50 lbs. a ton be the tractive resistance for the road and $4\frac{3}{4}$ H.P. per ton, or 6 H.P. for 25 cwts., is the horse-power required for this speed. If the car has a front area of 12 square feet the additional horse-power required for wind resistance is about $3\frac{1}{2}$; making in all $9\frac{1}{2}$ horse-power. This is the actual resistance at the road wheels; hence the engine power must be more than this to allow for the loss in transmission. Allowing 10 per cent. for friction, $10\frac{1}{2}$ horse-power is needed when the revolutions of the engine will give a speed of 35 miles an hour. The ratio of gear must be adjusted in accordance with this.

Fig 10 gives a table showing the number of revolutions required with any ratio of gear at any speed and for any sized wheel. It is calculated from the formula—

$$\text{Revolutions} = \frac{\text{Speed in miles per hour} \times 5820 \times \text{ratio of gear}}{60 \times \text{circumference of wheels in feet}}.$$

If the engine is to make 1,100 revolutions per minute at 35 miles per hour, and the car has 34-inch wheels, the gear ratio required is $3\frac{1}{4}$ to 1.

Tractive Resistance on Hills.—There is, generally, no difficulty in running the car at any required speed on the flat, but if any hills have to be ascended allowance must be made for the increased road resistance, which is ascertained by dividing the weight of the car by the gradient. Thus a gradient of 1 in 12 will increase the resistance by a twelfth of the weight of the car.

Putting this in graphic form, if the grades are expressed in percentages the curve becomes a straight line. Fig. 11 gives such a curve from which the resistance can be easily read. It shows both the resistance due to gradient in lbs. per ton and the total resistance, including various road resistances.

For instance, if with a $3\frac{1}{4}$ to 1 gear on its top speed the above car is to ascend a gradient of 1 in 15 at 35 miles an hour, we find (1) that the tractive resistance is 200 lbs. per ton and the horse-power required about 19 per ton or 24 for a 25-cwt. car; (2) the wind resistance will increase this to $27\frac{1}{2}$ horse-power; and (3) if 10 per cent. is allowed for friction an engine of 31 horse-power at 1,100 revolutions will run up the hill at this speed. If lower speeds are desired the calculation is the same, but the low gear *must* be low enough to ensure the car being able to ascend the steepest hill on the route.

A simpler way, perhaps, is to ascertain the tractive force by means of the following formula—

$$\text{Tractive force} = \frac{\text{Cylinder area} \times \text{stroke} \times \text{mean pressure} \times \text{ratio of gear} \times \text{efficiency of transmission}}{\text{Circumference of wheels} \times 2}.$$

The dimensions of cylinders and wheels being in inches; the pressure, in lbs. per square inch. The tractive force thus found divided by the weight of the car in tons gives the tractive force per ton.

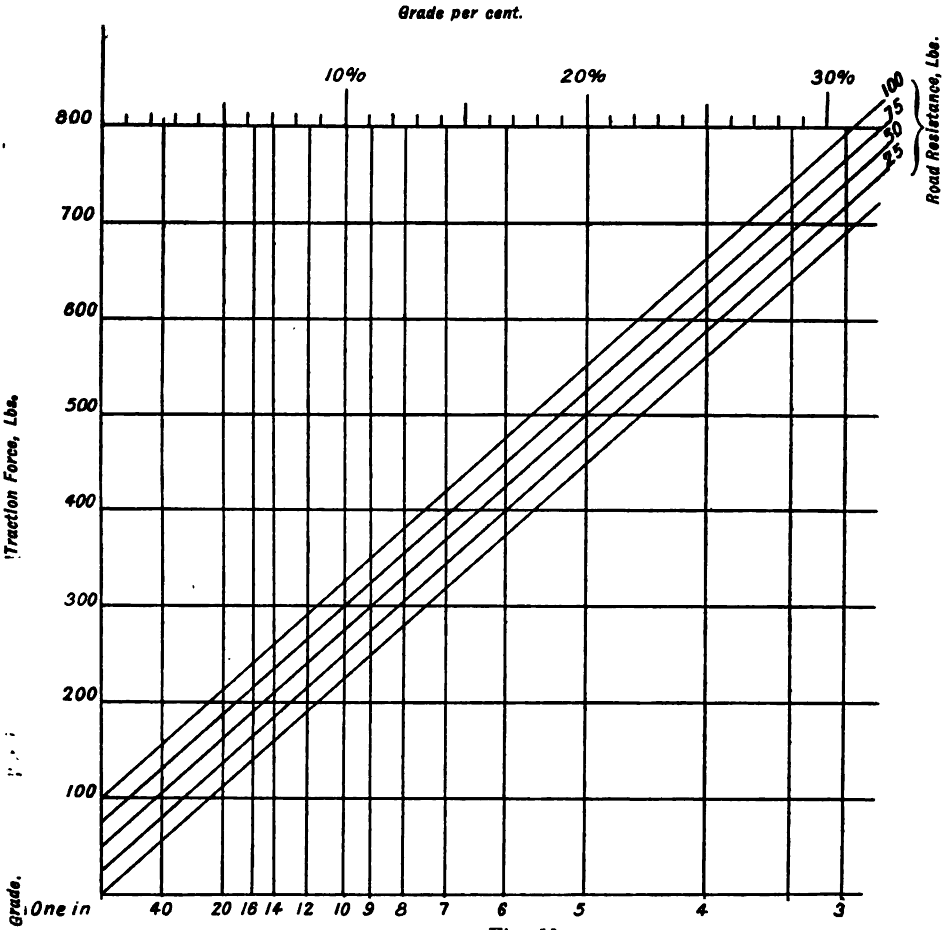


Fig. 11.

Directions for estimating the mean pressure and the friction in the transmission of power will be given in Chapters iii. and xiii. respectively. Perhaps the most important point in these calculations is to provide for the car having a sufficiently low gear for ascending the steepest hill likely to be met with. Many cars have been built which, although they have ample power, can only run in a country that is comparatively flat owing to

their gear being insufficiently low. It is owing to this defect that it is so often found difficult or impossible for cars to ascend a steep hill, to the great annoyance of the occupants. A modern car takes most ordinary hills on the second speed, so that the first is not often used, and there is no objection whatever to its being made sufficiently low for any ordinary gradient. For comparing the tractive forces required for various cars the mean pressure may be estimated at 80 lbs. per ton, while 10 per cent. may be allowed for the friction of each gear drive and 5 per cent. for that of a chain. The mean pressure and the friction will generally be less than these amounts, but the results agree well with the observed facts.

Taking this estimate as a basis, and comparing it with the actual performance of a car, it is found that a calculated tractive force of about 500 lbs. per ton will generally be sufficient, although with the steeper hills and when the engine is not in its best condition some of the passengers may have to walk. If the tractive force is increased to 600 lbs. per ton on its low gear a tour may be made comfortably through any ordinary country.

If the data are available the actual pressure shown on the brake and the actual friction of the transmission gear are preferable to any assumption. Cars should be made for use on all roads considered available for horses. Such roads frequently include short gradients of 1 in 5 with a very bad surface and a sharp turn in the middle.

Cylinder Capacity and Weight.—The most satisfactory results are obtained when the ratio of cylinder capacity to car weight is high, as the speeds required are then obtainable with fewer changes of gear and without overtaxing the powers of the engine. In most cases modern cars are amply fast enough on the flat; in fact, many of them are never run at their full speed. The gain of speed, therefore, must be obtained by increasing the speed up the hills and by a quicker resumption of full speed after slowing down for traffic, corners, &c. The larger cylinder capacity provides the reserve power necessary for these purposes, for emergencies, and for the easy running of the engine. The cylinder capacities of cars have varied from about 3 cubic inches per cwt. in some of the older cars to about 50 in some of the latest racers. The actual capacity per cwt. of the cars in several of the public trials are given in various tables and also the capacity of some of the modern cars from the makers' lists. From these it will be seen that the average is about 10 when fully loaded. It is evident that in making improvements an increase in this ratio will be of great importance.

CHAPTER III.

ENGINES—GENERAL ARRANGEMENT.

Motor.—The motor that propels a car is essentially the ordinary gas engine, in which an explosive mixture of gas and air is drawn into the cylinder on the outstroke of the piston, compressed to about a third or fourth of its original volume by the return of the piston, and fired (generally by an electric spark), the explosion driving out the piston; the exhaust valve opens as the piston begins to return, and the exhaust gases are driven out. The piston, therefore, only does useful work during one stroke out of every four, and the flywheel actuates the engine during the other three strokes. There is no great objection to this for fixed gas engines, as a heavy flywheel of large diameter can be used to keep the engine going; such heavy flywheels are inapplicable for most cars, and therefore they are generally fitted with several cylinders, so that the action may be uniformly maintained. It will also be seen that such an engine cannot exert any power unless it is first turning round; that is to say, it cannot start of itself against a load. In this it entirely differs from a steam engine, which can exert as great, or even a greater, pressure when not running than it can when running. In a steam car the engine can always be connected with the driving gear and start the car; but, if a petrol engine could not be disconnected from the driving gear, the car would have to be started by pushing it till the engine began to work. This is often done in bicycles, but is obviously not practicable for a car. The engine in this is so arranged that it can be connected with the driving gear by a friction clutch. The engine is first started with a handle, and then the clutch is put in so as gradually to start the car without stopping the engine.

Relative Power of Motor and Ordinary Gas Engines.—The present-day motor develops in proportion to its size several times as much power as the ordinary Otto cycle gas engine; yet it is only a gas engine with the proportions modified. For instance, a single cylindered Dion engine has a 4-inch cylinder and a 4¾-inch stroke for 8 horse-power. A stationary gas engine for the same power will have a cylinder about 8 by 12 inches. The one engine will weigh 1½ cwts., the other nearly a ton.

The essential point that makes this difference is that the motor can run at a much higher speed than the stationary engine. In all engines the power will increase in proportion to the speed up to a certain point, which differs from various causes in different engines. The main limiting points in most engines are, first, that the gas cannot get in or out of the cylinder quick enough; and, second, that the friction becomes so great that the engine cannot be run cool. The increase in speed may also cause such an increase in internal friction that it absorbs all the extra power produced. This is what actually happens in most small motors.

Valve Area—Lightness of Moving Parts.—The first difficulty is removed by making the valve area large enough, and the others by making the moving parts light. In fact, it may be said that the modern motor

depends entirely for its success on the lightness of its parts. A de Dion piston for an 8-horse engine weighs 3 lbs., and the top end of the connecting-rod half a pound. For a stationary gas engine these weights would approximate half a cwt. If the piston is heavy, there is great friction at high speeds, and, as it has to be started and stopped twice every revolution, the power of the engine is decreased and the wear of the brasses increased. The great improvement in modern motors is mainly due to making the engines with light moving parts and capable of running at high speeds. True, they are still often called slow-speed engines, but are always run fast when it is desired to get the best results out of them.

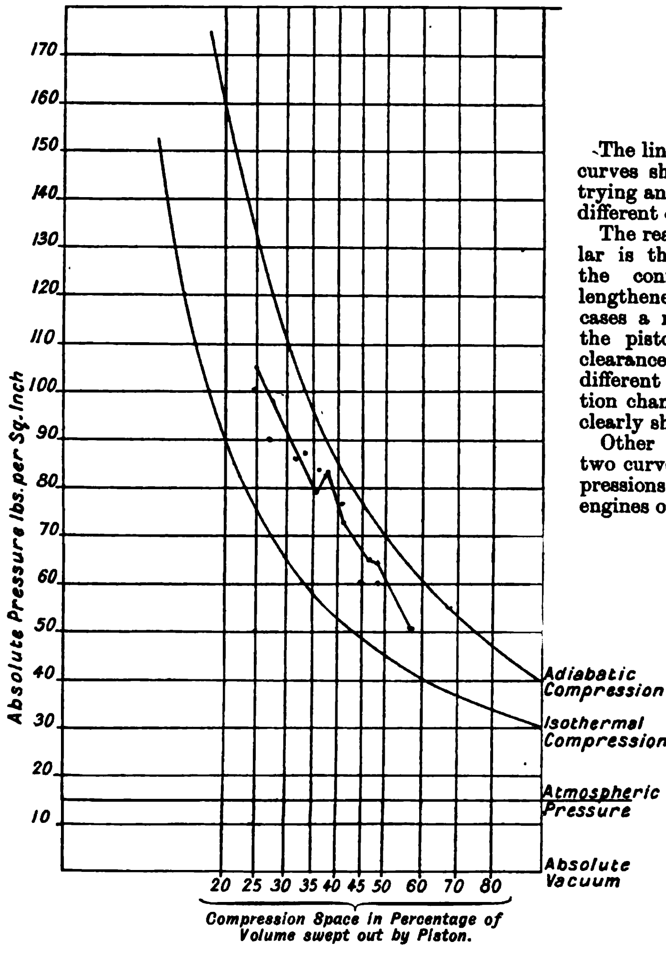
Ignition.—Another important factor is the ignition. This has to be as instantaneous as possible, as otherwise the explosion will not take place quick enough to develop its power before the piston has moved a considerable part of its stroke. Consequently, tube ignition, which has been so successful in fixed engines, is little used for motor work, and electric ignition of some kind is always used. This point will be dealt with further in Chapter iv.

High Compression Necessary for High Speed.—The compression to which the charge is subjected has also a good deal to do with the speed at which the engine will run. If the compression is very low, the charge will burn much slower than if it is rather higher; consequently, it may not explode quickly enough to develop its power during the working stroke at high speeds. In fact, the effect of a low compression is much the same as that of too feeble an ignition. Consequently, if we have a very powerful ignition, it will partly compensate for a somewhat low compression; while, on the other hand, a weak ignition will neutralise the advantages of a high compression.

In discussing the matter of compression, it is as well to consider it in terms of the ratio of volume of compression space to volume swept out by the piston, rather than, as is usually done, in terms of the pressure that is obtained at the end of the compression stroke. This is easily obtained in a gas engine, but not so in petrol motors; for these, indicator diagrams are not often taken—the power on the friction brake being the only item available. It is, however, quite easy to measure the compression space and the volume swept out by the piston; the ratio between these will give what is wanted. Thus, if the volume of the compression space is 20 cubic inches and the area of the piston multiplied by its stroke is 60, we have a compression ratio of 33 per cent.

The actual compression ratio to be used in an engine is one of the most important points in its design, and depends on many things. Taking, however, the point we are immediately discussing:—Developing the maximum power of an engine—it may be taken for granted that, up to a certain point, the smaller the compression space, and, consequently, the greater the compression pressure, the greater will be the power obtained. In the early gas engines the compression space was about 65 per cent. of the volume swept out, corresponding to a pressure of about 30 lbs. Later, the compression space was gradually reduced, and now it is about 25 per cent., the corresponding pressure being about 100 lbs. As the compression is raised, there is an increase in the explosive pressure, and, consequently, in the power developed from a given-sized cylinder, and also greater economy. The latter is, of course, the important point in large gas engines, whereas in motors it is the power.

There is a limit to the compression that can be usefully employed. In gas engines, for instance, an excessive compression pressure causes the charge to explode before the end of the compression stroke; this produces a back pressure tending to drive the engine the wrong way, and to put great strains on the working parts. In petrol engines there seem to be other limits as well.



The line between the two curves shows the results of trying an actual engine with different compression ratios. The reason it is so irregular is that in some cases the connecting-rod was lengthened and in some cases a ring was added to the piston to reduce the clearance. The effect of different shape of combustion chamber due to this is clearly shown.

Other spots between the two curves are actual compressions from gas and oil engines of various kinds.

Fig. 12.

Compression Curves.—Before going into the exact limits of useful compression it may be as well to look at what happens when a gas is compressed. If a gas is compressed at a uniform temperature the pressure increases in accordance with the formula PV = constant. This, as a matter of fact, never takes place in the cylinder of a gas engine. When a gas is compressed it becomes warmer in consequence of the amount of

work done in compressing it, and, therefore, the pressure rises considerably faster than the proportion given above, which is called "Isothermal" compression. If, on the other hand, the gas could be compressed in a cylinder which is absolutely a non-conductor of heat all the heat generated by the compression would be retained in the gas, and the compression curve would follow the law of "Adiabatic" compression. The exact ratio of increase in a gas compressed in this way varies with the specific heat of the gases; but fig. 12 gives a diagram calculated for the compression of air, showing both the isothermal and adiabatic curves, and also the curve taken from the experiments on a particular gas engine tried with different compressions by the Institute of Mechanical Engineers. It will be seen that, in this case, the actual observed compression pressure in all cases lies between the two curves. The reason of this is obvious, as when the gas is compressed, although the cooling effect of the cylinder walls will not keep it down to its original pressure, it will absorb some of the heat, and, therefore, prevent its compression increasing in proportion to the adiabatic curve.

Effect of Circumstances on Compression Curve.—How near the actual compression will approach the one or the other curve will depend on many things. In the first place, the temperature of the walls of the compression space will naturally have a great effect. The cooler these are the nearer the pressure will approach the isothermal curve. The time taken in compressing it will also have a very great effect. The shorter this time is (that is to say, the higher the number of revolutions the engine runs) the less heat will be lost to the walls. On the other hand, the greater the proportion of surface to volume in the compression space the more heat will be lost. Thus, a large engine has much less surface in proportion to volume than a small one, as the surface only increases as the square of the linear dimensions, while the volume goes up as the cube. Various differently shaped combustion chambers also offer very different amounts of surface in proportion to their volumes, and this is a matter which will be dealt with further in Chapter vi.

In stationary gas engines economy is the main point, and great care is taken in designing the combustion chamber to get the best results. Further, special means are now generally taken to clear out the whole of the exhaust gases from the combustion space at the end of each stroke by passing a charge of pure air through it between the exhaust and suction strokes. In petrol motors, on the other hand, the shape of the combustion chamber is usually determined entirely by the mechanical design of the engine, and no special scavenging arrangements are possible.

Compression Ratios in Practice.—In practice the ratios of compression space in use by different makers of petrol motors seem to lie between 30 per cent. and 45 per cent. The former is the ratio used by some of the most successful makers of high-speed engines, and probably represents the ratio which gives the best power for a given cylinder. Engines have been made with compression spaces down to 22 per cent. of the volume swept out, but, in making it as low as this, there seems to be a very considerable loss. What the exact reason of this is is not quite clear, but it is evident that, if the compression pressure of an engine is increased beyond a certain amount, there is a very considerable loss of power. In theory, a higher compression should be utilisable in a high-speed engine than in a low-speed one, but practice does not always accord with this.

Causes of Loss of Power with High Compression.—Several causes may account for this loss of power at high compressions. One is that the loss of heat through the cylinder walls is probably much greater. As the compression pressure is increased and the compression space reduced the proportion of surface to volume becomes larger, and generally the shape of the combustion chamber much less suitable. Also, as the gas is at a higher pressure, it will part with its heat much more readily to the cylinder walls. It is quite possible that these causes prevent the explosion developing its full pressure.

Practical Disadvantages of High Compression.—There are many other practical objections to very high compressions. The effect of any leakage is much more important at high pressures than low. The friction of the engine itself, as also the wear and tear, are intensified.

In practical work the compression is often governed by quite other considerations than that of developing the greatest power. As the compression is increased the inequality of the turning moment of the engine gets greater. An engine with a low compression, therefore, runs smoother

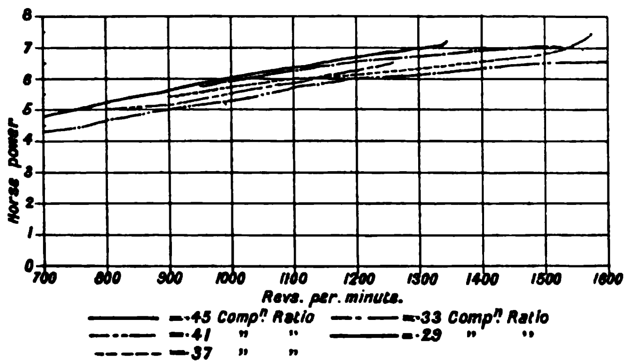


Fig. 13.

than one with a high compression pressure. This will be obvious when it is considered that the pressure at the beginning of the stroke is lower and is developed less suddenly, and that the compression pressure is a negative one, tending to prevent the shaft turning. In order to make engines run smoothly makers often use less compression than formerly. Therefore the requirements of different engines vary a good deal. If it is required to make the engine as small as possible for its horse-power a pretty high compression will be used, while if it is of more importance to make the engine run quietly and smoothly a lower one will be preferred.

In practice the loss of power by moderating the compression is not as great as might be imagined, provided a powerful ignition is used. Fig. 13 shows the results of some experiments on an engine with different ratios of compression. It was a three-cylinder engine with cylinders $4\frac{1}{4}$ by $4\frac{1}{4}$ inches, and only one cylinder was used, the others being removed. The results are therefore not so good as would have been the case if it had been a single-cylinder engine, as there were a good many parts that had to be driven. It will be seen, however, that the actual loss of power was not very great, even for a very large increase in the compression space. In practice,

for ordinary pleasure car work, the extra comfort obtained by lowering the compression quite compensated for the slight loss of power. In these trials ignition was by trembling coil only.

Speed Limits.—As has been seen, the essential point in the success of a petrol engine is that it should be able to run at high speed. Further, in practice the limit of speed at which the engine will run, and give satisfactory results, depends almost entirely on the lightness of its moving parts. Therefore, in racing engines, every effort is made to secure lightness; thus it is not uncommon to drill holes all over the lower part of the piston to lighten it. By straining every nerve in this way it is possible to make engines that will run satisfactorily at very high revolutions. For this purpose engines of 6-inch stroke are sometimes run up to 1,500 or even 2,000 revolutions a minute. These speeds are, however, beyond those at which engines, as ordinarily constructed, will run satisfactorily. Expressing the speed of an engine in terms of piston speed—*i.e.*, the stroke multiplied by twice the revolutions—it is not usual to run engines over about 1,000 feet piston speed a minute. In any particular engine there will be some rate of revolution at which it will give its best horse-power. As a rule, the turning moment will steadily diminish from a fairly low rate of revolution to the maximum at which the engine will run; but the revolutions will, for a time, increase faster than the turning moment lessens, and as long as they do so the horse-power also will increase. Fig. 14 shows the actual curve both of brake load

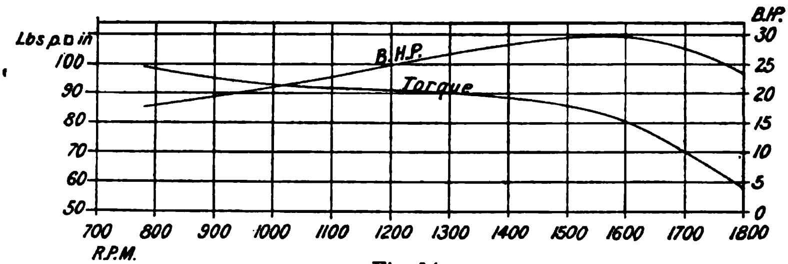


Fig. 14.

The figures on the left-hand side give the torque in terms of the mean pressure in the cylinder, no allowance being made for friction.

and horse-power obtained with the three-cylinder engine mentioned above at different rates of revolution. In this case the compression space was 29 per cent. of the volume swept out by the piston, and the engine was run with two ignitions—viz., (1) trembling coil, and (2) low tension make and break. It was noticeable that with either of these alone the power was considerably less than with the two together. The piston and top half of the connecting-rods with crosshead pin, &c., weighed, in this case, 4½ lbs. It will be seen that it gave the highest horse-power at about 1,500 revolutions, corresponding to a piston speed of 1,050 feet per minute.

Many of the older low-speed engines having comparatively heavy moving parts absorbed the greater part of their power in internal friction at such a speed as this. Consequently, even at moderate speeds, there was considerable wear and tear.

Revolutions Fixed by Practice.—The revolutions are often fixed by quite other considerations than the development of the best power of the engine. In the first place, moderate revolutions mean, as a rule, less noise,

and this is a very important point. Then, it is often not convenient to employ more than a certain ratio of gearing, and this, at a given speed, fixes the number of revolutions. In ordinary work there are probably few of the engines of modern high-powered cars which are run in daily work much over 600 feet of piston speed per minute. Still, it seems that it should be right to make an engine that can be run at high speed, if necessary, as then we get the greatest range of speed.

Calculating the Power.—In order to be able to calculate the power obtainable from a given engine we require to know the revolutions made, the mean pressure in the cylinders, as also the size and number of the latter. In order, therefore, to settle what sized cylinders are required for a given power at a given number of revolutions, all we require to know is the mean pressure obtainable. As mentioned above, it is usual to take this from actual indicator diagrams in stationary engine practice, and to allow what is considered necessary for internal friction. This is because it is more convenient in large engines to take indicator diagrams than brake tests. In petrol motors the usual plan is to take brake tests. It will be more convenient, therefore, to take the pressure as shown in the cylinder from the brake tests, and to neglect friction. For this reason the brake load in the diagram (fig. 14) is shown in terms of mean pressure in the cylinder. The mean pressure naturally varies a good deal with the construction of the engine and the compression pressure, &c. Generally speaking, it seems that higher mean pressures can be obtained with petrol than with coal gas. Under favourable circumstances pressures over 90 lbs. per square inch have been shown on brake tests, and no allowance being made for friction; and if this is allowed for at the rate of 20 per cent. it would mean an actual mean pressure in the cylinder of nearly 120 lbs. In order to get this, however, a somewhat high compression has to be employed, and in ordinary car work it is probably unusual to get pressures of over 70 to 80 lbs. The formula for estimating the power of an engine is, therefore,

$$\frac{\text{Total area of cylinders} \times \text{stroke} \times \text{mean pressure} \times \text{revolutions}}{33{,}000 \times 2}.$$

The area of cylinders is in square inches and the stroke in feet, pressure in lbs. per square inch, and the revolutions in number per minute. If there is more than one cylinder the total area is simply added together. The reason why the stroke is halved is that there is only one working stroke in every alternate revolution.

Taking 75 lbs. mean pressure and 650 feet piston speed per minute as a mean of what is about usual, this will give $2\frac{3}{4}$ square inches piston area per horse-power.

In comparing this formula with the rated powers of actual engines it should be remembered that some engines are rated at their maximum power and some at their ordinary working power. This is done for purely commercial reasons, and has nothing to do with the actual powers exerted by the engine, as can be demonstrated by comparing their horse-powers as shown by their performances in actual hill-climbing tests.

Proportion Between Bore and Stroke.—A point on which there is great divergence in design is the proportion of bore to stroke. Engines have been made in which the stroke varied from two-thirds of the bore to one and a half times. In engines for cars there is a general tendency to make the stroke just a little longer than the bore, but on racing cars and in racing

launches it is generally made less than the stroke, and on bicycle engines about the same. In theory at a certain piston speed the power should be independent of the stroke. On the other hand, the number of revolutions increases as the stroke is shortened, but there is a limit beyond which this is not practicable, as the same piston speed cannot be maintained with a very short stroke as with a long stroke. Assuming a certain piston speed, the shorter the stroke the lighter the engine will be per horse-power, but the higher will be the number of revolutions, and, consequently, the greater will be the wear and tear. On the other hand, if the number of revolutions is constant, the power will depend on the cylinder volume, and not on the proportion of bore to stroke, while the weight of the engine will not vary much per horse-power.

As a matter of fact, it is easier to maintain a constant piston speed than a constant number of revolutions. Theoretically, this would be most advantageous for the short-stroke engine. In practice this is true up to a certain point, but, as the stroke gets shorter, the number of revolutions per minute and the wear and tear increase. In fact, all the theoretical advantages are with the short-stroke engine and the practical ones with the long stroke. In motor cars and fast launches the weight of the motor is a matter of great importance. In a stationary engine, on the other hand, weight is of secondary, while durability is of prime importance. Hence, for motor car work, a comparatively short-stroke engine seems best, while, for stationary work, a long stroke is always used. For racing work of all kinds the shortest stroke that can be worked satisfactorily would be best.

The general conclusion is that a cylinder with the diameter and stroke equal is very satisfactory. It has the advantage that all the engines of a set can have the same proportion of diameter to stroke and all the dimensions in round numbers, whereas many engines are made with trifling, but disadvantageous, differences of dimensions for no reason whatever.

Some firms make the smaller engines with a longer stroke in proportion to the bore than the larger ones. I cannot quite see the object of this. In all other engineering it has been found that the right proportion of bore to stroke is the same for engines of all sizes (3 inches to 10 feet cylinders) if used for the same purposes. If no great variation of proportion is required for steam engines of this great range it does not seem that there need be any variation in proportions in petrol engines which have a much smaller range of size. No doubt, for special work, such as purely racing work, it is better to use a somewhat shorter stroke than in cars constructed for durability, but this applies to engines of all sizes. In any case the sizes of cylinders should be regulated by some definite system.

Valves.—The size of the valves is another important point in general design. The general tendency has been, and is, to increase the valve area. In the early engines the rate of flow through the valves was 10,000 feet a minute. Makers have gradually altered their patterns, and enlarged these, till the rate of flow has been reduced to much less than half this. Probably about 5,000 feet is the highest rate of flow that can be considered satisfactory. This means that if the piston speed is to be 1,000 feet the area through the valve must be a fifth of the area of the cylinder. Few engines exceed this, except temporarily. Assuming that the lift of the valve is a fifth of its diameter, the diameter of the valves should be half that of the cylinder. Few engines have valves as big as this, but I have used them on tricycle engines with good results. On the other hand, few engines run continuously

at 1,000 feet piston speed. Probably valves about ·4 of the diameter of the cylinder will serve all ordinary purposes.

Advantages of Large Valves.—In practice the advantages of large over small valves are, first, they require grinding less frequently, probably owing to the less cutting action of the slower rate of flow. Secondly, they need not be opened and closed so suddenly. The small valve, in order to get the gas in or out of the cylinder, has to be full open during as much of the stroke as possible. In fact, the exhaust valve has sometimes been so small that the engine has run best when it opened a long way before the end of the working stroke. Some have even opened not much more than half way through it. A valve of the right size will open just before the end of the working stroke (say 5 per cent. of the stroke). By the end of the stroke it will be nearly full open, and, as the piston is then hardly moving, it will be full open in ample time to let out the exhaust as the piston returns. When the piston is moving at half stroke on the return, and the gas has to be cleared out as fast as possible, the valve is full open. As the valve begins to close, the piston is slowing down, so that there is not so much gas to be removed. When the valve is small and the piston travels fast there is only a partial escape of the gas during the piston stroke; the liberation of the rest necessitates the valve opening suddenly, and remaining open to the end of the exhaust stroke, and to its being subjected to extreme wear and tear. It will also be much noisier, and one of the great improvements in modern motors is to make the cams of a form that will ensure quiet action, and if the valves are big enough this will not reduce the power given.

The extensive use of mechanically-worked inlet valves, called M.O.I.V., testifies to their merits. The mechanical valve can be made to open wider, as it has a strong spring to shut it, and is opened with a cam, and, therefore, is not liable to stick. It is suitable for all engine speeds, and the engine will give good power results whether it runs slow or fast; but the valve is more expensive to make than others, as it has several extra parts.

The automatic valve has to be made very light indeed to get good power out of the engine at high speed, and is, consequently, apt to break. Still, it has several advantages. It is very simple and cheap, and it enables the engine to be governed on the exhaust. This cannot be done with the mechanical valve, as, if the cylinder is not clear of exhaust gas when the inlet is opened, the gas flows through the carburettor. This method of governing keeps the compression constant and favours economy, and also, perhaps, will stop the oil working up into the cylinder quite as much as it does when the engine is throttled.

The inlet and exhaust valves are usually made of the same size in order to allow of their being interchangeable, and this is a rational plan, as all the gas which enters the cylinder must be discharged from it. The former practice of combining enormous inlet valves with very small exhaust valves has almost ceased.

Cylinders.—The number selected will depend on circumstances. The simplest, and generally the most compact, engines have a single cylinder, and are the most economical and much the cheapest to build; but the turning moment is very irregular, particularly at low speeds, owing to the necessity of making the flywheel small. The balance of advantages over disadvantages is such that the single-cylinder engine will probably long continue in use, especially where weight is not important, while if anything goes wrong there is no difficulty in having to decide which cylinder

is faulty. Fig. 15 shows approximately the turning moment of a single-cylinder engine, the effect of the flywheel being neglected.

If two cylinders are used the disadvantages are less. Several arrangements may be used, but the usual plan is to put the cylinders vertically side by side, and to work them on one crank, this gives a much more even turning moment, as indicated in fig. 16; but the moving parts cannot be accurately balanced. If the cranks are placed opposite, the moving parts are

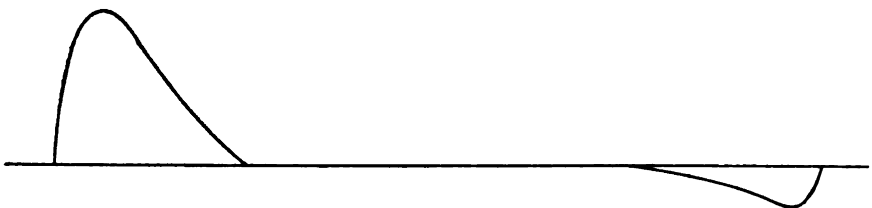


Fig. 15.—Diagram of Turning Moment with a Single Cylinder.

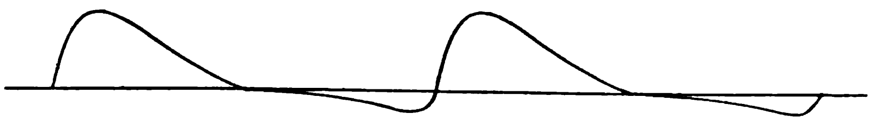


Fig. 16.—Diagram of Turning Moment with the Two-Cylinder Cranks together.

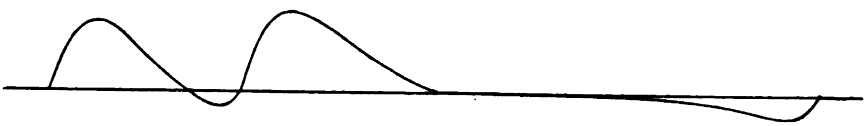


Fig. 17.—Diagram of Turning Moment with the Two-Cylinder Cranks opposite.

Fig. 18.—Diagram of Turning Moment with Two Diagonal Cylinders.

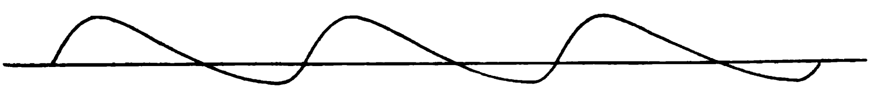


Fig. 19.—Diagram of Turning Moment with Three Cylinders.

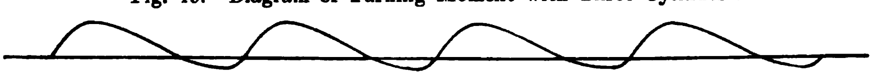


Fig. 20.—Diagram of Turning Moment with Four Cylinders.

better balanced, but the turning moment is not so good (see fig. 17), and the advantage over the single-cylinder car is hardly appreciable. By placing the cylinders diagonally on one crank the turning moment is a great deal better than in the last case, and the moving parts are also better balanced (fig. 18).

With three cylinders the explosions are evenly divided and the moving parts fairly well balanced. The turning moment is shown in fig. 19.

With four cylinders the moving parts are, for all practical purposes, perfectly balanced and the explosions are evenly divided, as indicated in fig. 20. In this, with a well-designed engine, we get a very even turning

moment and smooth running engine, and this will probably be the standard number of cylinders for ordinary cars in the immediate future.

By further increasing the number of the cylinders there is a still more even turning moment, but at the expense of considerably increased complication, each additional cylinder giving a proportionally less gain. On the other hand, for an equal power the cylinders become continually smaller and the parts more delicate. In practice the four-cylinder engine if properly made gives a turning moment sufficiently even for all ordinary purposes, the flywheel taking up the small unevenness there is.

The diagrams are all drawn for the same power, consequently the amplitude lessens with the increase in the number of cylinders. For this reason, as well as from the increased number of impulses, the flywheel required for a multicylinder engine can be made lighter the more cylinders there are. This also means that the whole engine can be made lighter for the same power, as the weight of the engine is chiefly due to that of the flywheel. In theory, when six cylinders are used there is no need of a flywheel, but the makers of six-cylinder cars fit them with flywheels as large as those employed for four-cylinder engines.

It must be borne in mind that, as the strength of the impulse is inversely proportional to the number of cylinders, the transmission gear must be made heavier for transmitting the power from one cylinder than from several. For instance, one 6-inch, two 4½-inch, or four 3-inch cylinders would give the same power at the same piston speed.

General Arrangements of Engines.—With regard to the general arrangement of the engine this depends very largely on the arrangement of the valves in the cylinder, which will be dealt with under the head of cylinder design. At present the usual plan is to put the valves opposite and this produces a general arrangement somewhat like fig. 21. This may be varied by placing all the valves on one side, and driving them all off one shaft or by putting one valve over the other, which makes an arrangement something like fig. 22. The cylinders may also be cast either all separate, in pairs, or all together. In the former and last cases it is usual to have the cylinders at even distances and to have a bearing between each crank. When in pairs the cylinders are generally arranged so as to get the centres of each adjacent pair as close to each other as possible and no intermediate bearing, as shown. The advantages and disadvantages will be discussed at length under the heading of cylinders and crank shafts.

A somewhat different arrangement to either of these is that in which all the valves are on the top of the cylinders. This is shown in fig. 23. This has the advantage that it gives the best shaped combustion chamber from theoretical considerations, but it is not quite so convenient for arranging other things. These three general arrangements will give a fair idea of the usual ones adopted. It will be quite understood that it is not necessary to cast the cylinders all in one piece when the valves are at the top of the cylinders, or to have them in pairs when they are at opposite sides. The changes of cylinders, all cast in one or in pairs or all separate, can be rung with any arrangement of valves, &c.

Importance of Simple General Arrangement.—Simplicity of design is of prime importance. Generally speaking, provision must be made for (1) cam shafts to work the valves (in some arrangements there are two of these, in some only one); (2) various auxiliaries, such as a pump and a magneto, and in some cases a pump for the oil where this has forced

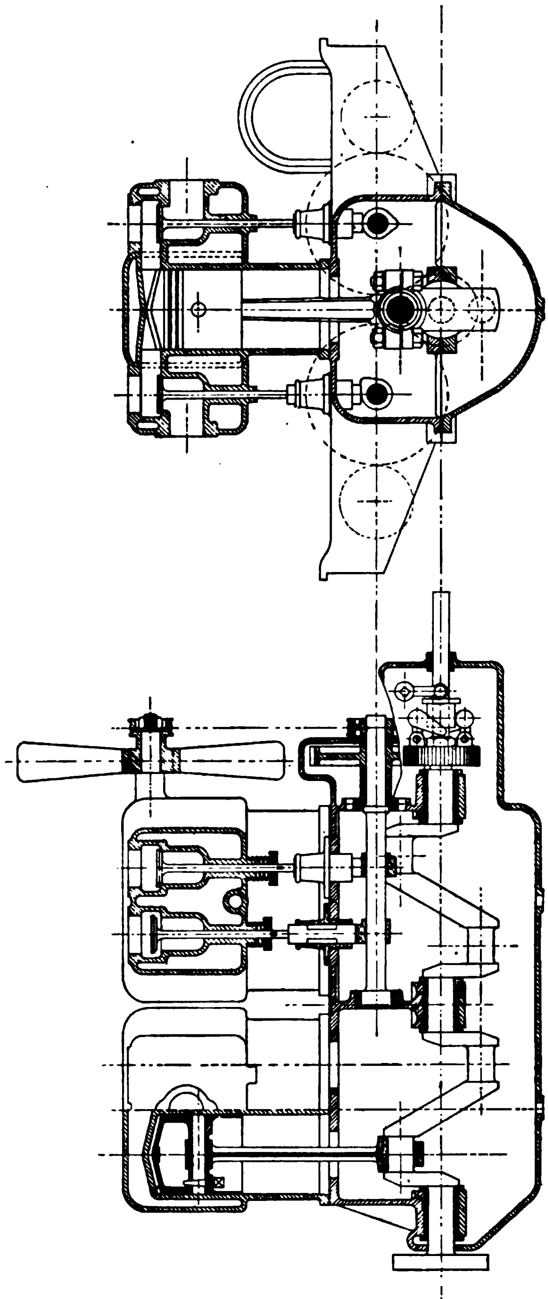


Fig. 21.

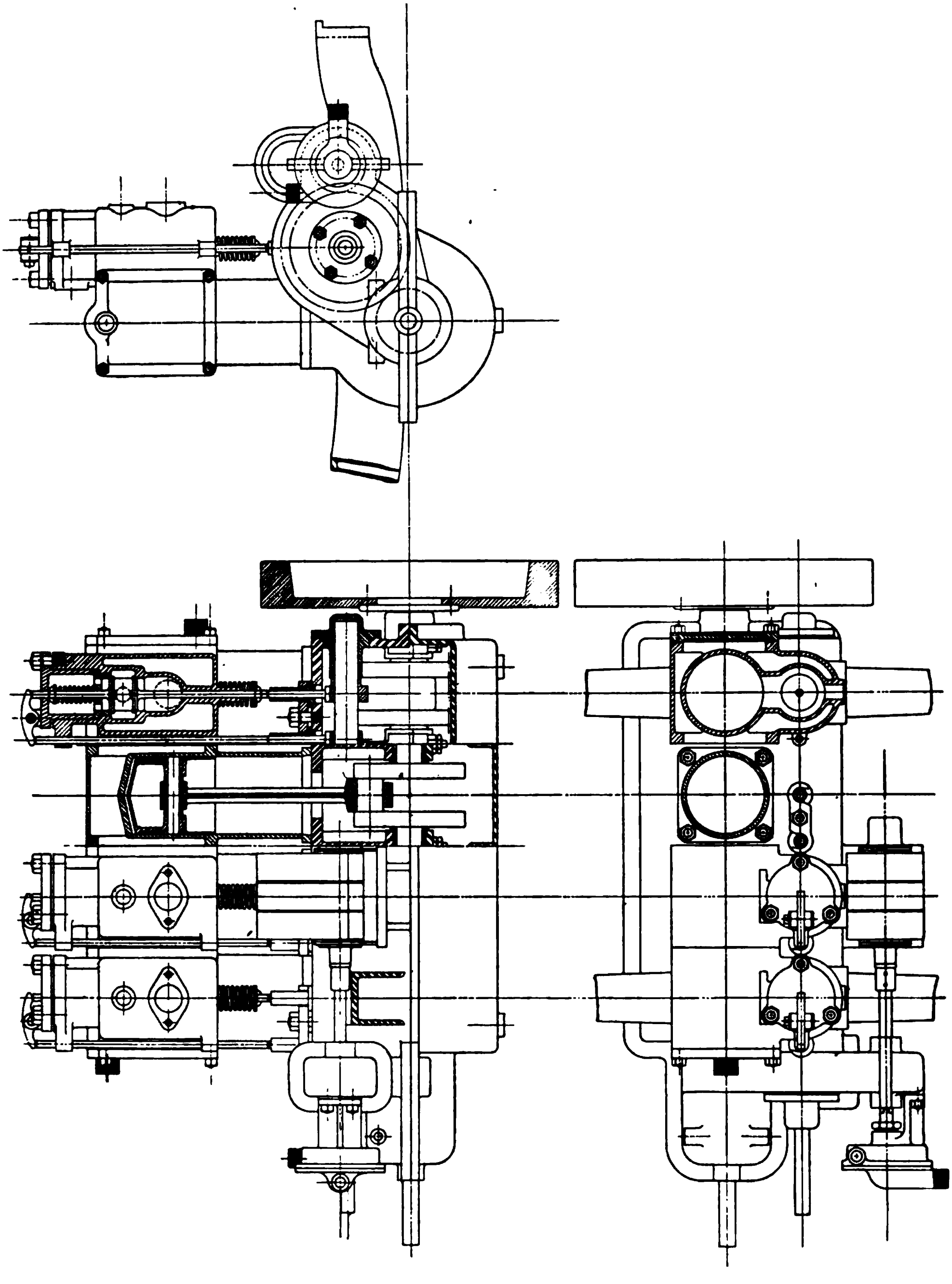


Fig. 22.

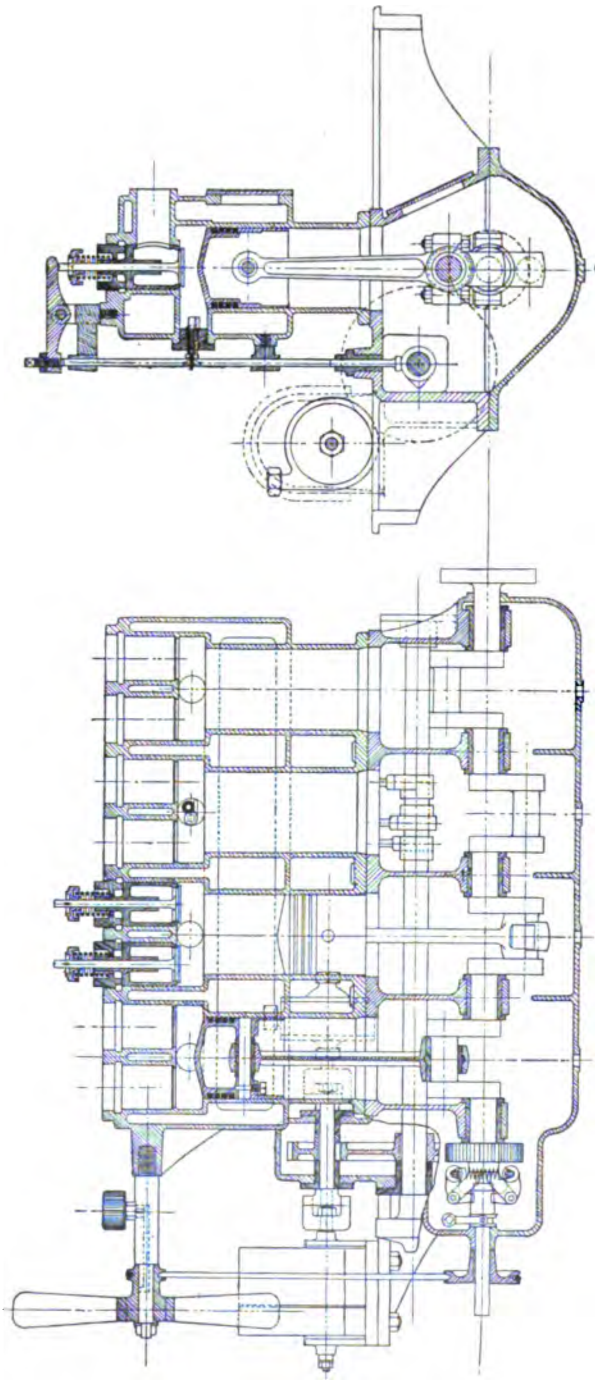


Fig. 23.

circulation. Where both high-tension and magneto ignition are used there must also be a commutator for the low-tension current, and with some magnetos there must be a distributor for the high tension. In arranging these as few counter shafts and gear wheels as possible should be used, and care taken that the engine is easily accessible.

Generally speaking, it should be possible to have the magneto and pump on one counter shaft and to drive this with one pinion gearing into the cam shaft pinion. Figs. 24 to 29 show various arrangements in actual use for driving these parts, but there are many others.

Fig. 24 makes a very neat arrangement, but necessitates a special magneto being made, as in those on the market the shaft does not come through. Fig. 25 avoids this, but it places the pump in a less convenient position than others. It should be possible, however, to make a quite satisfactory arrangement of this. Another plan is to reverse the arrangement of fig. 24 and take the spindle through the pump instead of the magneto (fig. 26). In this case a special pump is needed, which is much less inconvenient to make than a special magneto. In fig. 27 both the pump and magneto are placed in convenient positions, but the extra gear wheel adds to the expense. In fig. 28, a common arrangement where there are two cam shafts, the only objection is the great number of gear wheels, while the special counter shafts for the pump and magneto make it expensive. Sometimes parts are driven with a chain, as in fig. 29, but this seems hardly so satisfactory as driving them all with gear.

Fig. 30 makes a very neat arrangement when the height of the counter shaft is sufficient. When this is not the case, the pump and magneto are often driven from a cam shaft in much the same position, but driven from separate skew gear.

Governor.—The governor is usually mounted on the engine shaft, where it runs the full speed of the engine. It is, however, sometimes mounted on the end of one of the cam shafts, in which case it only goes half as fast, and therefore must be more powerful and heavier to give the same effect.

Fan.—A fan is generally placed behind the radiator, in order to induce a draught when the car is standing still or going with the wind; this necessitates a pulley being placed on either the crank shaft or cam shaft from which to drive it. There are advantages in both. If the pulley is placed on the crank shaft, the ratio of the pulley wheels will not need to be so great as if placed on the cam shaft, as the latter goes slower. On the other hand, the governor projects beyond the end of the crank case proper, and therefore, if the pulley is placed on the crank case, it will be several inches further forward than if placed on the cam shaft (compare figs. 21 and 23). This means that the fan, as also the radiator, must be placed further forward, thereby adding to both the length and the weight of the bonnet. It is better to place the fan on the engine than on the radiator, because the latter arrangement requires the radiator to be erected with the fan pulleys being in line, while the strain of the belt on the fan may cause leakage in the radiator. If placed on the engine, the parts, being machined, will be in line without fitting, and the cost will be less.

Commutator.—If high-tension ignition is used, a commutator is required to distribute the low-tension current to the coils, and a high-tension distributor may also be needed. In the latter case the two are generally combined. This is most cheaply placed at the end of the cam shaft, but with the common arrangement of radiator and bonnet would not be very

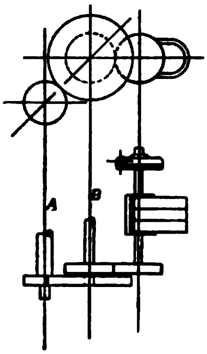


Fig. 24.

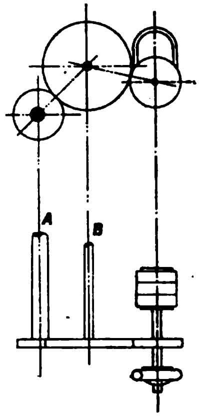


Fig. 25.

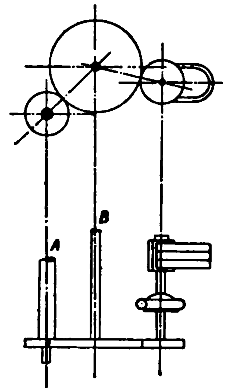


Fig. 26.

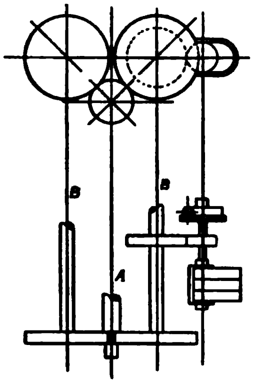


Fig. 27.

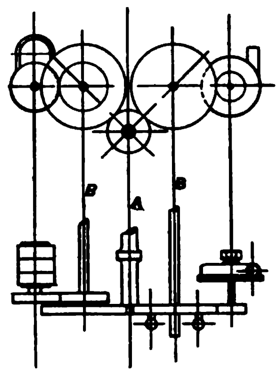


Fig. 28.

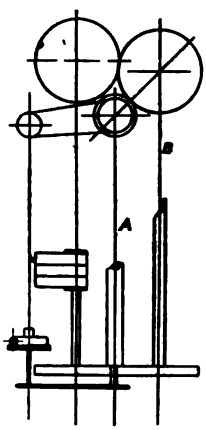


Fig. 29.

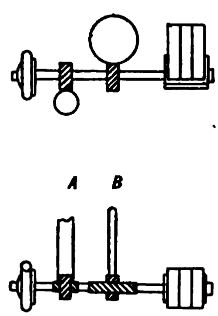


Fig. 30.

A, Crank shaft. B, Cam shaft.

accessible. This is an objection with most commutators sold. Consequently they are better put where they are within easy reach, usually at the side of the crank case, and driven with bevel or skew gear (fig. 31), which

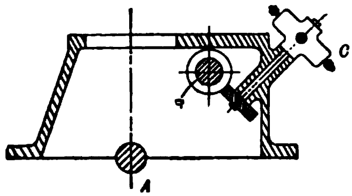


Fig. 31.

A, Crank shaft.
B, Cam shaft.
C, Commutator.

entails extra parts and, consequently, extra expense, both in manufacture and upkeep; sometimes on the top of the cylinder, being driven by a bevel wheel at the forward end of the cam shaft, which entails a special bracket on the cylinder as well as on the crank case, and therefore requires more machining and fitting up. With a suitably arranged radiator and bonnet, the best place is on the end of the cam shaft, or it may be put in front of the radiator and carried on an extended spindle, and it is then easily reached.

CHAPTER IV.

IGNITION.

Essential Conditions.—An American has said that a gas engine consisted of "an ignition with an engine built round it." He was not far off the mark, as it is certainly the soul of the engine.

The ignition of a fast-running oil engine must be simple and absolutely reliable, and with the shortest possible explosion. In many cases the timing should be adjustable.

Time of Ignition.—In slow-running stationary engines it is usual to arrange the ignition to ignite practically exactly at the end of the compression stroke. In this case the explosion is often fairly complete before the piston has started to move. In the case of high-speed engines this is not enough, as the time for explosion is so short that, if the ignition takes place at the top of the stroke, it does not develop till the piston is some way down the stroke. If the explosion was really instantaneous, it would be correct to ignite just at the end of the compression stroke under all circumstances. It takes a very perceptible time, as a matter of fact, and, in order to get good results, it is necessary to make the ignition point some way before the end of the compression stroke. In an engine running at 1,000 revolutions a minute, to get really good results the explosion ought to be complete in something like one-hundredth of a second. It is probably nothing like complete in this time. In many oil engines the explosion is not quite complete even at the end of the working stroke, as can easily be seen by running without the exhaust pipe, when flame is often seen issuing from the exhaust passage.

The time the explosion actually takes depends very much indeed on the igniting force itself. When the ignition is caused by a very small electric spark, it takes a much longer time to develop than when it is caused by a large spark or some large hot surface. The result is that, the more feeble the ignition, the longer must be the interval of time it is applied before the end of the stroke to allow of the explosion occurring at the right moment. If the ignition is too feeble, even this precaution will not make it satisfactory.

Fig. 32 represents a theoretically perfect diagram. This assumes that the ignition is exactly at the end of the stroke and that explosion is instantaneous. In practice this is never carried out, but any failure to do so results in the rounding-off of the corners of the diagram and loss of power. In actual stationary engines, diagrams can be got which are very close indeed to the theoretical, as can be seen from fig. 33. In this case the ignition causes a sufficiently instantaneous explosion to get the ignition line practically vertical. Suppose we have an engine running so fast and with such an ignition that it must be advanced 25 per cent. of the stroke to get the best result, our diagram will be something like fig. 34. The loss of power is obvious, as the ignition begins before the end of the compression stroke and, therefore, increases the back pressure there. It is also not complete till after the beginning of the working stroke, so we lose pressure

here. Therefore, the more powerful the ignition and the less it is advanced the better.

There is another preferential reason. If the ignition was instantaneous, it would be right to ignite at the exact end of the stroke for every speed, and there would be no need to adjust the time of ignition for different speeds. Although in practice this precision is unattainable, it is quite possible for the ignition to be so good that it wants *very little* adjustment indeed to obtain the best result for different speeds. That is to say, the engine can be run at all the usual speeds without varying the ignition point, and results got which are sufficiently satisfactory. This simplifies matters very much, both in arranging the control levers and in driving.

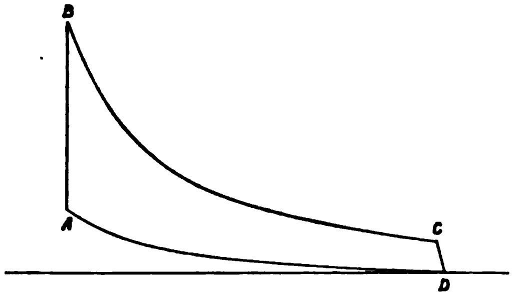


Fig. 32.

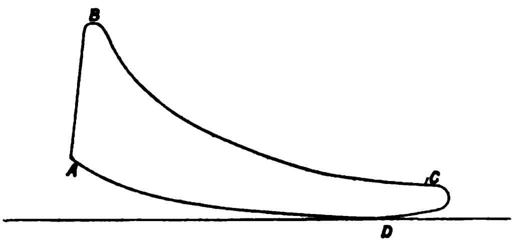


Fig. 33.

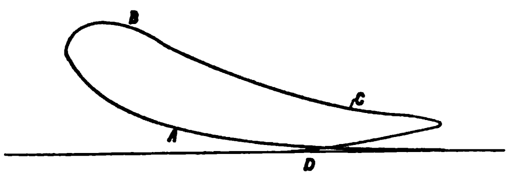


Fig. 34.

FIGS. 32, 33, AND 34.

A, Ignition point.
B, Point where explosion is practically complete.
C, Point where exhaust valve opens.
D, Point where pressure in cylinder has fallen to atmosphere.

If, with such a weak ignition as indicated in fig. 34, we advance the ignition to get the explosion developed earlier, and then give more forward pressure, we get more back pressure on compression stroke. If we put it later to avoid the excessive back pressure from A to end of stroke, we get a still more delayed explosion. The effect of having too small an exhaust valve is shown by the back pressure from C to D.

A powerful ignition has the further advantage that the engine is less dependent on the precise character of the mixture of gas and air, and will act with a larger variation in their ratio. In all machinery that is to work satisfactorily there must be some margin; that is, the parts are made stronger than necessary to allow for possible contingencies. If, in driving steam engines, the boiler can only supply just enough steam when all the conditions are favourable, there will be a deficiency of steam whenever the conditions are less favourable. Consequently, the boilers are made capable

of supplying sufficient steam for working the engines under all ordinary conditions. Similarly, a feeble ignition will fail if everything is not adjusted to a nicety.

Various Means of Ignition.—The various means of ignition that have been used at different times for internal combustion engines are as follows, taking them approximately in order of their introduction:—

1st. *High-tension Electricity.*—This was used by Lenoir in his early gas engines in 1860, as also by Priestman in his first practical heavy oil engine, but is now rarely used for stationary engines owing to its unreliability.

2nd. *Flame.*—The flame passes through a small slide valve into the cylinder and ignites the charge. This is almost obsolete.

3rd. *Tube Ignition.*—This is very largely used for stationary engines.

4th. *Automatic Ignition.*—This is largely adopted in stationary oil engines using heavy oil.

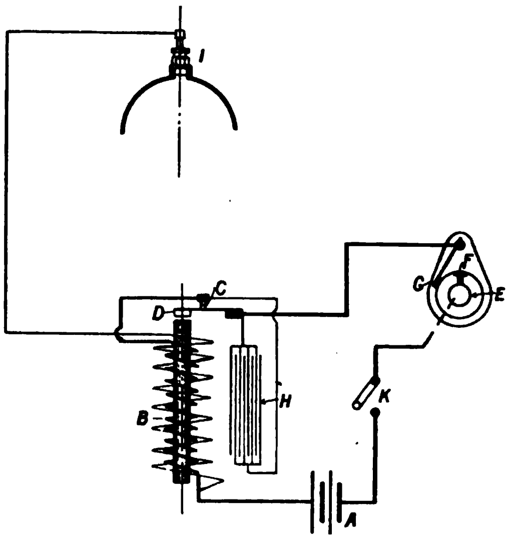


Fig. 35.

A, Battery or accumulator.
B, Induction coil, consisting of a primary coil of a few turns of coarse wire, and a secondary, consisting of a great many turns of very fine wire.
C, Contact breaker where circuit is broken by the core attracting the trembler, D.
E, Commutator on end of cam shaft, which closes circuit at the right time by bringing the metal segment, F, against the brush, G.
H, Condenser, to minimise sparking at contact point, C.
I, Ignition plug, where secondary current is taken inside the cylinder and jumps across a small gap to earth. The other end of the secondary circuit is joined to the primary coil and earthed with it at the commutator, E.

5th. *Low-tension Electricity.*—This method is employed in many stationary engines, and almost exclusively in marine petrol engines.

Of these, the high-tension electric, tube, and low-tension electric are the only ones extensively used in motors. The tube is rarely used for pleasure cars.

High-tension Electricity.—The high-tension electricity system is the oldest form of ignition, and is of French origin. For stationary work it has been abandoned by the vast majority of engine-makers in favour of either tube or low-tension. It has also been long since abandoned by the American marine motor-makers.

Fig. 35 shows the general arrangement of the ordinary high-tension ignition, as used on motor cars, which is practically the same as in the Priestman and Lenoir engines. In this there is a battery or accumulator, generally the latter, giving four or six volts; a contact breaker, to close the current at the time that ignition is required; and an ordinary induction

coil. The high-tension current from this is led to the cylinder, where it passes through an insulated plug and makes a spark across two points in the cylinder. The spark is very small, the points usually being set less than the sixteenth of an inch apart, and often half this.

The coil has the usual automatic trembler, so that a succession of sparks is kept up as long as the contact is closed at the contact breaker.

Difficulties.—This type of ignition can be made to work very well, but has several disadvantages. The insulation of the high-tension circuit is always a difficulty, as the current is very small and its tension very high, often several thousand volts, which renders it very liable to be short-circuited—*e.g.*, by damp or oil, by defects in the insulators (porcelain or mica), or by coatings of soot or oil inside the cylinder.

The induction coil, if well made, gives no trouble, but some of those supplied for motor cars are so insufficiently insulated that they do not last long. The trembler in some coils requires constant adjustment to make them work well, and the commutator often fails. Owing to these failures this form of ignition is chiefly confined to cars, as in these the failures are not of serious importance and are easily remedied. Such failures are much more serious in stationary engines, as a stoppage of the driving engine of a factory involves much inconvenience and loss of time. It is much the same with marine engines. Here a stoppage may be very serious and, in fact, may easily lead to the loss of the boat and perhaps her crew. Consequently, in the United States of America, where there are thousands of marine motors in actual use for many purposes in all weathers, low-tension ignition is almost universal. As freedom from stoppages on the road from any cause, however small, is of the utmost importance, and it is probable that in the future the low-tension ignition, which has proved itself the most reliable so far, and which is certainly the simplest, will come into general use.

Defective Timing.—Another objectionable defect is the difficulty of timing the ignition, especially in high-speed engines. The trembler shown on the coil is of the ordinary make and break pattern. This works well enough for really slow-speed engines, but not for high. The reason is that there is a perceptible interval between the closing of the circuit and the first spark. This will be evident from the nature of the action of the make and break. The first spark will occur when the first break takes place, and this is not until the core has been sufficiently magnetised to attract the armature. Thus, the first spark will not take place till some time after the contact is closed. If the engine runs at a constant speed and if this interval is constant the contact must be set some way ahead of the time the ignition is required, and the ignition point will then be always the same. It is, however, quite unlikely that the interval will be *exactly* the same twice running, and, therefore, the time of firing will not be the same. This will entirely spoil the performance of the engine. Another very serious disadvantage is that, as the interval will be about the same at any speed of the engine, the point of ignition will be later as the engine goes faster, whereas it should be earlier. This involves a constant alteration of the contact point either by hand, which is troublesome, or automatically, which complicates the machinery. The difficulties are partly overcome by the use of more quickly vibrating light tremblers. Each coil maker has his special pattern, but that depicted in fig. 36 will illustrate the principle, which, however, is not perfect. The trembler vibrates so fast that the core of the coil is never fully magnetised, and,

although the sparks are more rapid, they are individually smaller. To ensure perfect ignition the first spark should take place at absolutely the right time. With a high-speed coil it is probable that the charge is fired by a succession of small sparks, and that the explosion travels more slowly through the charge than one started by one big spark. At all events it must be wrong in principle, as if the first spark really fires the charge the others are unnecessary; and if several sparks are needed, as seems to be the case, the result will be more or less uncertain firing. Hence, it does

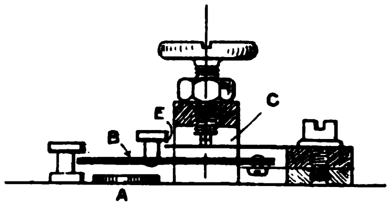


Fig. 36.

A, End of core of coil.
B, Trembler.
C, Contact point.

The trembler blade, B, is made very light, but has a little movement before it touches the contact breaker at E, and acquires enough momentum to break the contact.

not seem to be possible to get the same power out of an engine with a trembling coil as with one big spark.

The amount of adjustment required by tremblers varies so much that in purchasing coils this point needs careful consideration, as also does the speed at which the coil will respond, which is far more important than the rapidity of vibration after the first response, because, as has been pointed out, the promptness of the *first* effective spark is the essential factor. A rough test of how quickly the coil will respond is made by just brushing the wire from the accumulator against the terminal of the coil. This is not very accurate, but it can easily be demonstrated in this way that the

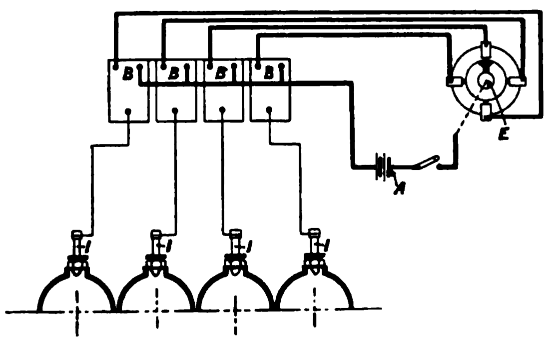


Fig. 37.

A, Battery or accumulator.
B, Coils, each has its condenser and contact breaker made up in the same case as coil.
E, Commutator, distributing current to the respective coils at right time.
I, I, I, I, Sparking plugs in cylinders.

quickness to respond in a coil does not necessarily correspond to the speed it will buzz at when once started.

With engines having more than one cylinder there are several variations of arrangement. Thus, in a four-cylindered engine we can have (1) a coil for each cylinder each with its own trembler; (2) a coil to each cylinder with one trembler to the four coils; and (3) one coil and a distributor for the secondary current.

Fig. 37 shows the wiring for the first arrangement. The difficulty is in so adjusting the tremblers that all will respond with equal speed. The

ignition must take place in each cylinder at exactly the same point in the stroke, and in order to insure this the first spark of each must occur at exactly the same time after the contact is made; probably this is never *absolutely* the case, but with careless adjustment of the tremblers the error may be considerable and the power of the engine will suffer accordingly.

This difficulty may be met by having only one trembler for all the coils. The connections for this are shown in fig. 38. This is a very convenient

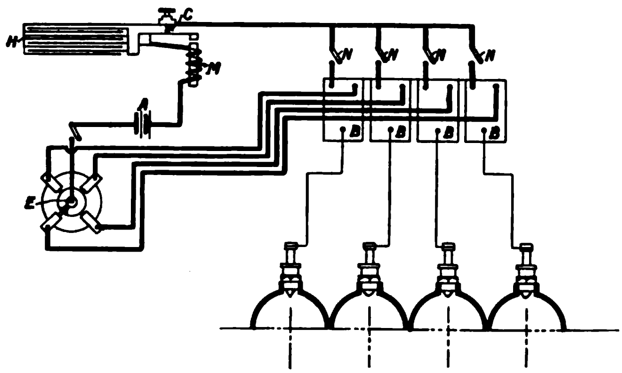


Fig. 38.

A, Battery or accumulator.
B, B, B, B, Coils.
C, Contact breaker worked by auxiliary coil, M.
E, Commutator.
H, Condenser.
N, Switches, to switch off each coil separately.

It will be understood that in practice all the four coils with the contact breaker and condenser are enclosed in one box.

system indeed and works well. In the best form of it there are actually two tremblers, only one of which is used at a time. This is a great practical advantage as adjusting a trembler on the road is troublesome, particularly if it happens to be dark and raining. With the double pattern there is a switch, so that either can be used at will. Then, if there is trouble with the trembler on the road, the second is switched on, and the first is left to be adjusted on reaching home. Coils of this pattern can also be

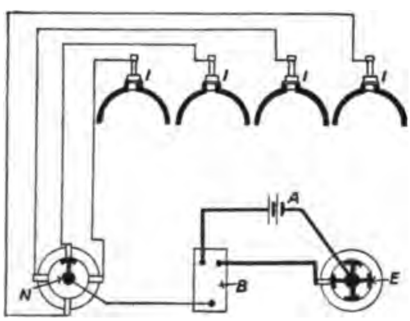


Fig. 39.

A, Battery or accumulator.
B, Coil.
E, Commutator, closing circuit four times each revolution of cam shaft.
N, High-tension distributor, distributing current to plugs, I, I, I, I.

In practice the commutator, E, and distributor, N, are generally mounted on the same shaft and are often combined in one.

conveniently made with a separate switch to the low-tension circuit of each coil so that it can be cut out. In this way the firing of each cylinder can be tested independently.

A third plan is to have one coil and a high-tension distributor. The wiring of this is shown in fig. 39. This also gets over the difficulty of adjusting several tremblers and does away with three of the coils, but it

introduces a distributor for the high-tension current. This allows of the system being much cheaper, lighter, and more compact, but it increases the chance of short circuit or a leakage through the distributor; hence a larger coil and accumulator may be required. The greater difficulty of putting cut-outs on the high-tension circuit is a drawback.

Single-contact Ignition.—There is an alternative high-tension ignition in which only a single spark is used. In this case, instead of using a trembler on the coil, there is a contact breaker on the engine, which makes the contact once every other revolution, and breaks it once. This gives one single spark to fire the charge. This is obviously right in principle, as the same amount of energy spent in one big spark will produce a much more definite ignition point than if spent in numerous small ones. It might, therefore, be expected that more power would be developed with this form of ignition, and this certainly seems to be the case. Although it is the fashion to look on this type as to a certain extent obsolete, it is still used by De Dion and several other of the leading makers, noted for the great power yielded by their engines. Fig. 40 shows the wiring for this form of ignition for a single

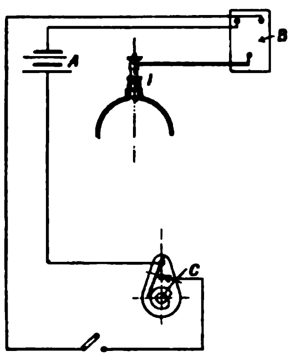


Fig. 40.

A, Battery.
B, Coil with no contact breaker.
C, Contact breaker on cam shaft.
I, Ignition plug.

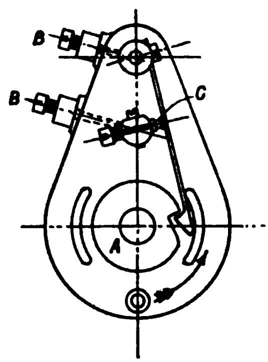


Fig. 41.

A, Cam on cam shaft.
B, B, Terminals.
C, Contact point.

cylinder, but for more than one it is the same, with either a coil to every cylinder or one coil and a high-tension distributor.

The success of this form of ignition depends a great deal on the contact breaker. Fig. 41 shows an early form, which works well only with springs of the right temper. If too hard, the spring breaks; and if not hard enough the pressure is too slight to give a good contact. It will also be observed that the distance the contact points part is very small indeed, and the parting consequently very slow. It must be remembered that it is at the break of the primary that the spark occurs, and that it depends on the break being instantaneous. The result of the slow parting is that the engine has to be turned at a good speed to make a spark at all, so that it is difficult to start. Also, the time of the spark is affected by the distance the spring drops into the notch in the cam. This means that with more than one cylinder there may be a lack of the adjustment required for it

to fire at the same point. Fig. 42 shows a variation used by several makers in which some of these points are remedied. The contact spring is pressed strongly against the point, and the break is generally a little more rapid than in the last arrangement. A further improvement is that the contact point is at the opposite side of the spring to the cam, and is thereby less liable to be oiled than in the fig. 41 arrangement, in which the oil may be thrown from the cam on to the space between the points, and so bring about the stoppage of the current. In the arrangement shown at fig. 43 (which works well) the break is very rapid, the points well parted, and the adjustment easy to make.

Where single contact is used, a dry battery is often used as a source of current instead of an accumulator. The coil, having no trembler, has very little internal resistance when the contact is closed, except the self induction. As long as the engine is running fast this will stop any great amount of current passing through, but should the engine stop with the contact closed there is nothing whatever to stop the current pouring through. Also, when

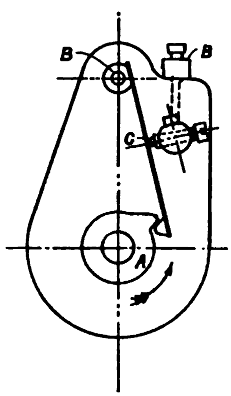


Fig. 42.

A, Cam on cam shaft.
B, B, Terminals.
C, Contact point.

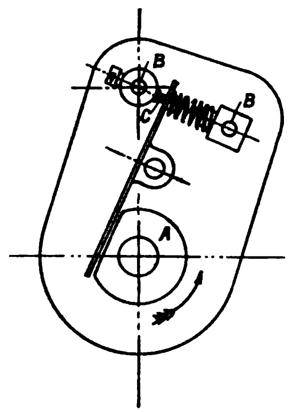


Fig. 43.

A, Cam.
B, B, Terminals.
C, Contact point.

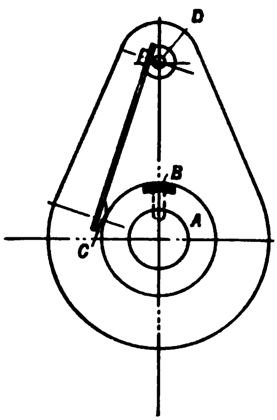
the engine is running slow the consumption may be rather large for the same reason. The dry battery has considerable internal resistance in itself, so that when the current reaches a certain amount this prevents an excessive amount going through. On the other hand, as there is one spark only, it does not require current for so long as the trembler coil does. For these reasons dry batteries that will run a single-contact coil of suitable construction for a good time will very soon run out with a trembling one, while accumulators will run the latter much longer than the former. This depends very much on the winding of the coil no doubt. Probably the two require different windings.

In the single contact one advantage is that the spark takes place at exactly the same place in the stroke whether the engine runs fast or slow, whereas with a trembling coil the lag causes the ignition to be later, as the engine runs faster. The result is that the single contact requires much less apparent advance than the trembler, and much less adjustment for different

speeds to get the best result. In fact, for all practical purposes one position for starting and one for running at all speeds is ample with a good spark.

Great difficulty was often found in the early engines with single-contact ignition in starting, for unless the contact breaker was very accurately adjusted, the spark was not good enough to allow of the start being made, but with the trembling coil there was a succession of sparks as long as the contact was closed. With properly wound coils and proper contact breakers, however, there is not much in this. On the other hand, the tremblers being dispensed with, there is no trouble in adjusting them.

A point in favour of the trembler is that it makes it much easier to locate a failure in the primary circuit. If the trembler buzzes all is right. There is no reason why an ignition should not be arranged with a trembler which could be switched in or out of the circuit, and which could be used for starting or testing, and switched out when running. In this case the trembler and the coil should each have its own condenser.



A, Cam made of vulcanised fibre with metal segment.
B, Metal segment, which is earthed to cam shaft.
C, Brush rubbing on cam.
D, Terminal for one wire, the other being earthed.

Where there is more than one cylinder there is a brush for each.

When B comes under C circuit is closed.

To alter time of ignition, the whole of the commutator except the cam is rotated round the shaft.

Fig. 44.

Taking it all round, if properly carried out, the single-contact system seems to give as little trouble and more power than the trembler. A great deal of the prejudice against it has been caused by employing unsuitable combinations of coils, contact breakers, dry batteries, and accumulators.

In all the above contact breakers it is of the greatest importance that the contact points should be of platinum, and of good size. As platinum is expensive, silver, and even nickel, are often substituted; or the platinum is made so thin that it quickly burns through. The result is that the contact is bad, and the ignition ceases to work properly. However well the coils are designed, there is always a little sparking at the contact point, and it is very largely the adoption of cheap contact points, particularly on "imitation" parts, which has given the single-contact system a bad name.

Commutator.—When trembling coils are used, there must be a commutator to distribute the current to the different coils. Theoretically, it should give no trouble, but in practice it gives a good deal, mainly owing to inappropriate manufacture, such as making the parts too slender and the screws too small. In the best makes now, however, these troubles are not common.

Any of the contact breakers shown for single-contact ignition can be used for trembling coils, but it is usual to use such as have a rubbing contact, not the end contact between the points as in the single-contact arrangement. The advantage of having a rubbing contact is that the surfaces are always kept clean, and there is consequently less resistance.

Fig. 44 shows a rubbing commutator of the simplest kind, which works well if properly proportioned, well protected from dirt, and properly lubricated. Its disadvantage is that as the cam and the brushes wear away the working is disturbed. The cam wears unevenly, as the fibre is softer than the brass; and, when worn, a little ridge is left at the edge of the brass, which causes the brush to jump as it goes over it, and, therefore, does not make a good contact. Moreover, the brushes wear away, and this alters the timing, as the more they wear the earlier will they make contact. If the wear was equal this would not be of consequence, but, as it may be uneven, the timing of the cylinders in a multi-cylinder engine becomes

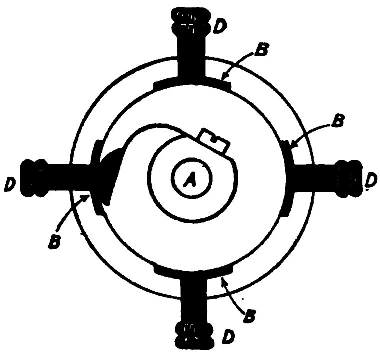


Fig. 45.

A, Cam shaft.
B, B, B, B, Segments for four cylinders.
D, D, D, D, Terminals.

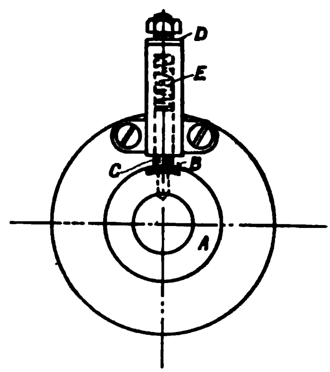


Fig. 46.

A, Cam.
B, Segment.
C, Wire gauze brush held up by spiral spring, E.
D, Terminal.

uneven. This objection may be avoided by having one spring on the revolving part, and segments for each cylinder in the casing, as shown in fig. 45. Gauze brushes, having a small guide and a spring to ensure close contact, as in fig. 46, have also been used with good results.

The use of a roller instead of a brush does not seem so good, as, should any dirt get on to the surface, it will not be wiped off, but rolled in. Also, there is an extra contact point for the current to traverse between the roller and the pin it runs on. It is also rather more expensive to make.

The earthed end of the wiring in both single contact and trembling coil ignition is often earthed to any convenient part of the frame of the car. The current then finds its way through the various parts of the frame and engine bearings to the contact breaker. This works well, but it would seem better practice to take the earthed wire right to the commutator, and have a proper contact there; this is now being done by several makers.

Where high speed trembling coils are used there seems no reason why the

contact segments should be as long as they often are. In the old days of low-speed trembling coils the arc of contact required to be something like an eighth of the circumference of the commutator in order to give the coil time to tremble. With a four-cylinder engine this means that the contact is closed for half the time, and, therefore, that the coils are taking half the current that they would take if run continuously. With coils that tremble fast this should be quite unnecessary. One eighth of the circumference of the commutator means half a stroke of the engine, and it is quite clear that it is not necessary to have sparks passing at the plug when the engine is anything like half way down the working stroke. If the charge is not well ignited by the time the piston gets to the top of the stroke arrangements should be made to ensure this, as the current should not be wasted in sending sparks across the plug long after the charge is well ignited. If, therefore, there is a short contact at the commutator, the current can be greatly economised, and a more powerful coil can be used, giving bigger sparks for the time it is actually running.

If one of two commutators closes the current twice as long as the other, not only will the coils require twice the current, and therefore the accumulators want charging twice as often, but the contact points on the coils will also want twice the attention.

Source of Current.—The current for the coils is usually furnished by an accumulator of two cells giving 4 volts. Many people, however, prefer to run with three cells giving 6 volts, a plan which has considerable advantages. When everything is exactly right the 4-volt accumulator does its work perfectly, but it has very little margin. In addition to the resistance of the wiring and coils themselves there is in the circuit the resistance of the contacts, both at the commutator and the trembler on the coil. The latter is only held in contact by a very light spring, and therefore the contact is not really very good. There is also the resistance of some sort of rubbing contact in the earthed circuit. It is not surprising that with all these breaks in the circuit a 4-volt accumulator should often fail to yield a current quickly enough for producing a spark at the right time. In fact, here, as elsewhere, there is need for a good working margin.

If the single contact system is used the number of breaks in the circuit is fewer, as we can connect each wire to the contact breaker itself. Moreover, the points of the contact breaker can be held together by a strong spring as they are parted mechanically.

When a single trembling coil and a high-tension distributor are used the commutator will, of course, be the same as the one for a single cylinder, but with a contact for each cylinder.

One point of importance is that the cam on the commutator should be so marked that, if removed, there can be no question as to where it should be replaced. Otherwise it is a very tedious thing to find the correct point, particularly on the road. Further, it requires fixing on the shaft in such a way as not to be likely to come loose; a single set screw is not sufficient.

Timing.—Timing the ignition is effected by moving the commutator slightly round the shaft, which alters the point at which contact is made. It is better to mount the body of the commutator on a spigot in the crank case than on the cam shaft itself, as, in the latter arrangement, the bush it is mounted on is liable to wear, with the result that the commutator is not rigidly held, and the contact is badly made.

Tube Ignition.—The next form of ignition in chronological order is the

tube. This is still used very largely on stationary engines, in which it almost superseded high-tension electricity. In its simplest form it is shown in fig. 47. The tube is open to the compression space, is heated by an external lamp, and is closed at the outer end. The action appears to be as follows:—At the end of the exhaust stroke the tube is full of burnt gases, and on the suction stroke the fresh ones do not mix with these to any great extent. On the compression stroke the fresh gases are driven up the tube by the compression, and, on reaching the hot part, are fired by the hot walls. The exact time of firing is, therefore, determined by the length of the tube and the position of the heating flame. It is also, to a certain extent, affected by the heat of the tube. In large gas engines there is usually a timing valve to let the gas into the tube at the right time, but this is a complication that seems unnecessary in small engines. At all events it is very seldom used in the smaller stationary gas engines, and these have cylinders far bigger than those of most car motors.

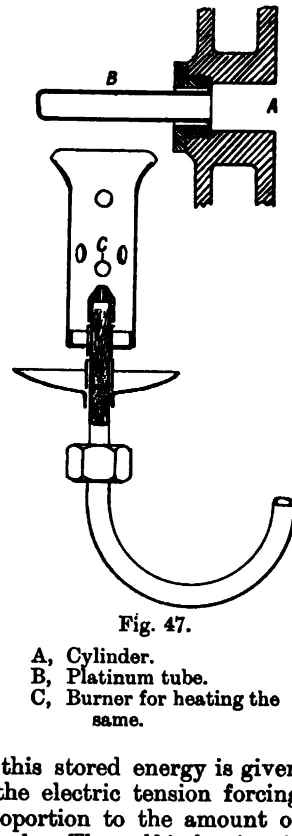


Fig. 47.

A, Cylinder.
B, Platinum tube.
C, Burner for heating the same.

Probably tube ignition would have been more successful in cars if it had not been for the trouble of the lamps. In gas engines these naturally give no trouble, as they are simply gas burners. In the cars, however, burners to burn petrol had to be used, and they were of the type of the plumber's blow-lamp, in which petrol is vaporised under slight pressure, and the vapour burnt through a small nozzle. They were troublesome to light in bad weather, and were rather liable to have the nozzle hole stopped up, as it was very small indeed, and required frequent inspection. Still, the system worked very well, and it is quite possible that it may be found suitable for commercial work where heavy oil is used.

Low-tension Electricity.—This is the latest form of ignition, and has come very largely into use. When a current of electricity is broken there is a spark. If the current traverses a long coiled circuit this spark is intensified. During the starting of the current electric and magnetic energy are stored up in proportion to the electrical capacity and the self-induction of the circuit. During the stoppage of the current this stored energy is given out again. If the stoppage be a sudden break, the electric tension forcing a spark to leap across an air-space is raised in proportion to the amount of energy so stored and to the quickness of the break. The self-induction is greatly increased by the presence of a soft iron core in the coil. For this reason there is always more or less trouble with the contact breakers in high-tension coils, and special arrangements have to be made to reduce this as much as possible.

Fig. 48 shows, diagrammatically, the principle of this form of ignition. Practically, it is the same as the low-tension part of the high-tension ignition system, with the high-tension winding, &c., left out—viz., the battery, the

low-tension coil and core, and the contact breaker. The current is furnished by a battery or accumulator of from 4 to 8 volts, and the coil consists of an iron core (laminated) some 6 inches long and 1 inch in diameter, with about a dozen layers of insulated wire on it. The wire is usually about No. 14 to 16 gauge.

In this ignition it is essential that the break in the cylinder be as quick as possible. In this case a very powerful spark can be produced; the quicker the break the larger the spark. It is quite easy to obtain a spark that will set fire to loose cotton (such as the covering of insulated wire unravelled) with a single spark. With a trembling coil it will not be done, unless a stream of sparks is allowed to play on it for a few seconds, which means probably several thousand sparks. There are no difficulties with insulation, as the voltage of all the circuit is quite low and the circuit is so simple that it is hardly possible that there can be faults in it which are not

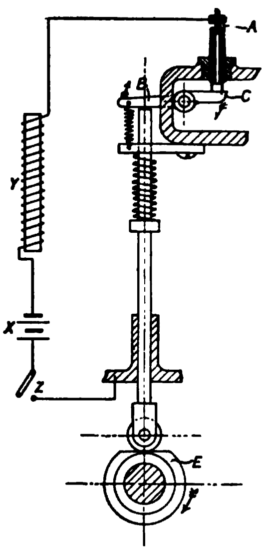


Fig. 48.

X, Battery.
Y, Coil.
Z, Switch.
A, Insulated plug in cylinder.
C, Contact point where circuit is closed and broken.
B, Lever worked by rod running on cam, E.
E, Cam on half-time shaft.

The cam, E, as it revolves, allows the lever, B, to be pressed down by the spring so as to close the contact at C for a short time, during which the current flows through the coil, Y; as the revolution continues the cam pushes the lever up, thereby breaking the contact and making a spark. The wire from Z is earthed to any convenient part of the engine.

obvious. The fault is generally in the contact breaker, and is easily detected by the naked eye. This form of ignition has been in extensive use in the United States for many years for stationary gas engines and for marine work, and is now being adopted in Europe for this class of work as also for motor work. In the latter, it is generally used in conjunction with a magneto machine to produce the current instead of a battery. This, however, makes no difference in the principle of working and will be referred to later.

Contact Breaker.—A good contact breaker is essential to the success of this ignition, but many makers have made it insufficiently strong for the work it has to do. The contact breaker should be as simple as possible and yet should give as rapid a break as possible. All wearing and striking parts should be of ample size and well made and fitted, and the surface of the plug inside the cylinder should be large enough not to burn away rapidly. The

break should be at a definite time and should not vary with the speed of the engine.

Trip Gears.—In most of the gears used in the United States the break is

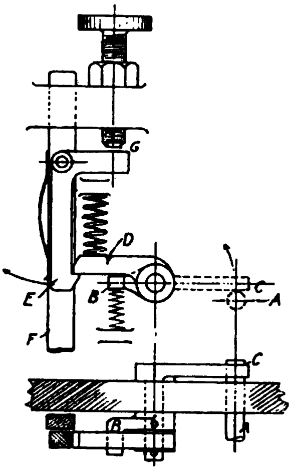


Fig. 49.

A, Terminal carried through cylinder wall on insulating materials.
B, Arm on ignition spindle, which goes through the cylinder wall and makes contact at C.
D, Hammer.
E, Lifter.
F, Rod carrying lifter.
G, Screw which trips lifter.

The rod carrying lifter is moved slowly up and down by a cam or eccentric on the cam shaft.

As it rises the lifter, E, catches the hammer, D, and lifts it; the arm, B, is pressed up by the spiral spring underneath it and the contact closed at C. When the lifter touches the stop, G, the lower part moves sideways and the hammer, D, drops sharply on B, and breaks circuit, the spring pressing D down being stronger than that pressing B up.

The time of ignition is altered by screwing G up or down.

made by some form of trip gear. There is a finger on the end of a spindle which passes through the cylinder wall and this finger touches the point of

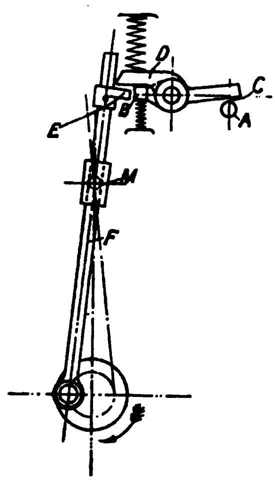


Fig. 50.

The action of this is the same as in fig. 49, the letters corresponding, except that the rod is pivoted at M, and moves laterally to trip the lifter, E.

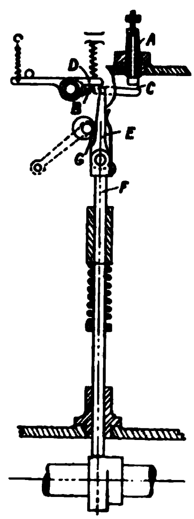


Fig. 51.

The action of this is the same as fig. 49, the letters corresponding.

The stop, G, is mounted eccentrically on its pivot so that when rotated it catches the shoulder on E, earlier or later.

one of the terminals. This is insulated where it goes through the cylinder. The other wire is earthed, and so the current passes through the spindle and finger to the wire. At the right time a spring loaded hammer is let drop on a lever at the end of the spindle and knocks it off the terminal. This makes a very rapid break.

Fig. 49 shows the arrangement used in one form of American launch engine. In this all the parts are very substantial and not liable to get out of order. The time of ignition can be regulated by moving the stop against which the bell crank presses to release the hammer. Fig. 50 shows another variation of much the same arrangement. There are innumerable slight variations of the same general principle.

Few examples of this type have been made in Europe. One is shown in fig. 51. Its mechanism is very much more delicate than in most of the American forms, and the surfaces hammered on are very small. The stop

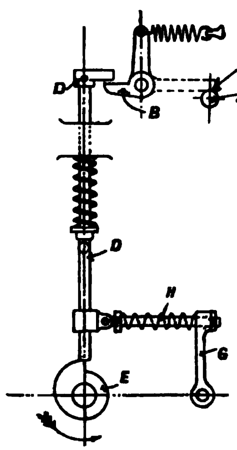


Fig. 52.

A, Insulated terminal.
B, Arm of ignition spindle.
C, Contact point.
D, D, Hammer which forms a rod going down to the cam shaft.
E, Cam which forms lifter.

When the cam, E, turns so that the rod, D, slips over the projection the latter falls and breaks contact.

The rod, D, is arranged so that it can be moved sideways by the arm, G, in order to alter the timing of the ignition.

It is held in place by the spring, H, so that if the engine turns backwards it moves against the spring and is not broken.

against which the lever trips to release the hammer is also so close to the axis that the slightest wear very materially disturbs the timing.

All these gears have the theoretical objection that there must be a certain amount of lag between the release of the hammer and the break inside the cylinder. If the hammer is light and the spring strong this is probably so slight that it is not of the least importance, and, as a matter of fact, such forms of ignition work perfectly well with no adjustment for timing.

The type favoured in Europe is that in which the cam forms the trip, and the general arrangement is that shown in fig. 52. It is somewhat simpler than the American gears, but the hammer parts are very much heavier; consequently, unless the spring is very strong the runner will not follow the cam closely; hence the ignition will be later the faster the engine goes. The defect of having such heavy moving parts for the hammer is a liability of their hammering themselves to bits. There must also be some means of arranging for the engine to go backwards without breaking anything. This is conveniently done by holding the runner in its place with a spring so that it can move out sideways when the engine accidentally turns backwards.

A variation, which does away with a great deal of the momentum of the parts, substitutes a rotatory movement of the rod for the longitudinal one, as shown in fig. 53. In this case the movement of the rod itself is so slight that its momentum is negligible, and that of the levers at the end only has to be dealt with.

In many cases the general arrangement is as in fig. 52, but the cam is made with a very steep side, not with a true trip, so that no special arrange-

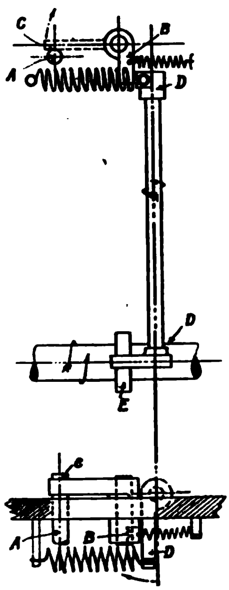


Fig. 53.

Lettering and action as in fig. 52.

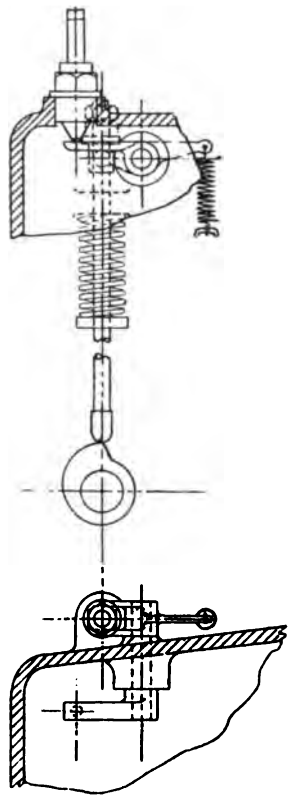


Fig. 54.

ment is needed for preventing damage if the engine goes backwards, as the runner can ascend the cam and can follow the cam sufficiently well to ensure that the time of ignition is the same at all speeds. This is illustrated in fig. 54. In this case the cam must be so proportioned that the break takes place quickly or the engine will not fire satisfactorily at low speeds. One point is that the cam must be of good diameter. The angle of the steep side will be the same for all diameters of the cam, but the arc traversed by the runner will vary inversely as the diameter of the cam, a remark which also applies to the positive break next described.

Positive Make and Break.—The best arrangement in many ways seems to be that in which the contact is broken by the cam direct. The object of the cam and trip gear is to get a rapid break, but if the parts are made light there seems no reason why the break should not be made positive, and yet be rapid enough for all practical purposes. This simplifies the arrangement, and ensures the ignition being at the same point for all speeds of the engine. One of the earliest forms of this ignition used in cars is essentially the same as that shown in fig. 48. This did not give a very rapid break, and the result was not altogether satisfactory, as the engine would not start with less than 16 volts, and the points burnt away very rapidly. Possibly the coils were not quite satisfactory, but as long as the

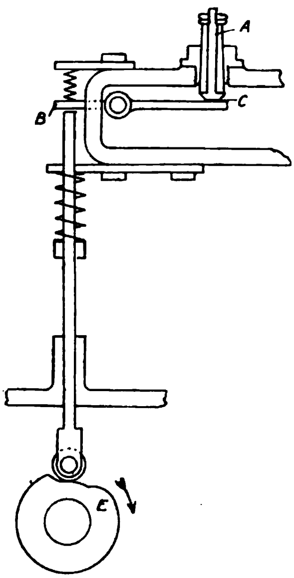


Fig. 55.

This is the same as in fig. 48, but the proportions of the arm, B, outside the cylinder and that inside are different, and the cam, E, is much steeper where it breaks contact.

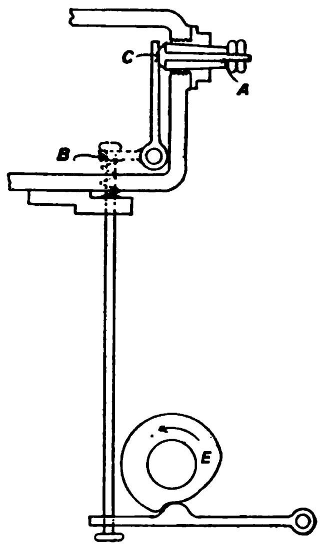


Fig. 56.

Letters as in fig. 52.

break is slow there will always be trouble, as an incipient arc is formed as the points separate. By simply modifying the proportions, as shown in fig. 55, we get a much more rapid break, and this runs all right.

In this type the moving parts must be very light, as, if they are heavy, their inertia will wear the cam very fast. It is possible to make them somewhat lighter if the rod is in tension than if it is in compression. This may be arranged as in fig. 56. In this type of gear it is even a greater advantage than in any other to have the rods rotating and not moving longitudinally. In this case the general arrangement will be much as fig. 53, but with a differently shaped cam. The simplest of all is to take the spindle straight on to the cam, as shown in fig. 57, which I have used

myself. The Singer bicycle also had a somewhat similar arrangement, and, as the engine was entirely enclosed in the wheel, and often covered with mud and wet, it is probable that few types of ignition would have worked there.

All the above are shown to work off the cam shaft, or from a point close to it. In some cases, however, the cam is brought to the ignition plug. This can be done with several of the trip gears, and simplifies things a good deal. Fig. 58 shows one form. The plain knock-off arrangement simplifies things, as shown in fig. 59. This is about as simple as an ignition could be.

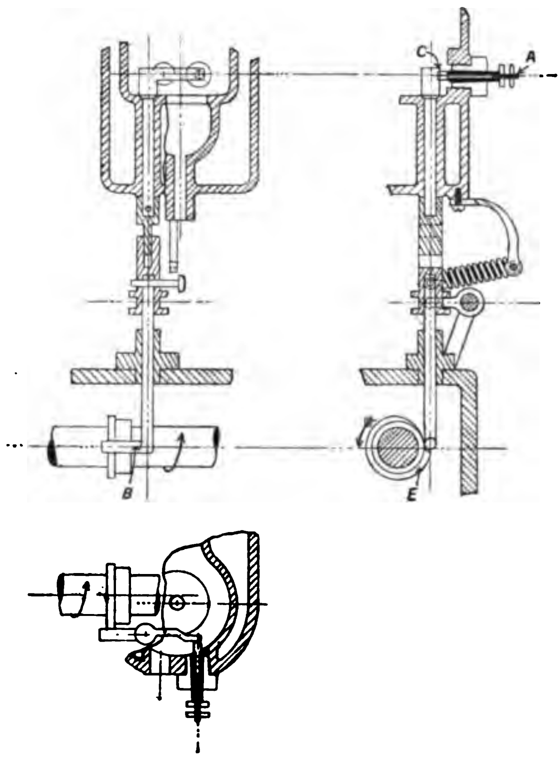


Fig. 57.

The arm, B, comes against cam, E, which allows the contact, C, to be closed at the right time by the spring. It is shown with contact just about to be broken.

In comparing these different gears it would seem that the advantage was entirely in favour of the plain knock off over any of the trip gears, provided the break can be made quick enough. In the latter there are generally two springs acting in opposite directions, and the contact depends on their relative strength. In fact, in many of the American type trip gears there are three or even four springs to each cylinder. Although springs are generally to be relied on, they do break sometimes, so that the fewer there are the better. In the knock-off type one only is required for each cylinder, and this can be strong, as the break is positive, and does not depend on another spring.

The cost of these arrangements varies much, but, as a rule, the simplest are the least expensive; complicated arrangements are, therefore, undesirable from every point of view. At the same time, the simplest of all the ignition arrangements, such as those shown in figs. 58 and 59, require the cam to be brought up to the ignition plug, and this means two more wheels and some bearings.

In the actual contact breaker inside the cylinder the small rocking arm is almost always earthed, and the other terminal taken through an insulated

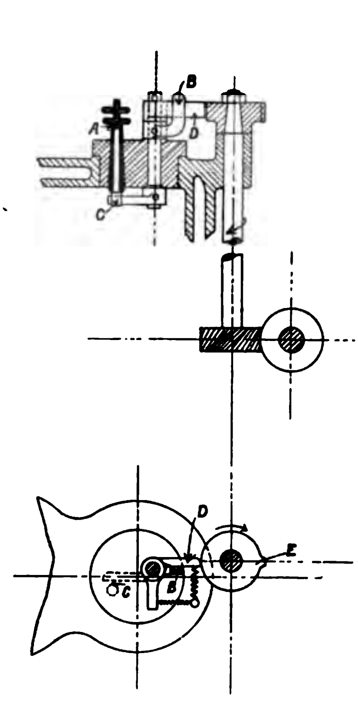


Fig. 58.

Letters as in fig. 49.

As the cam comes round, the projection, E, on it catches D, then, as it passes by, D drops sharply on B.

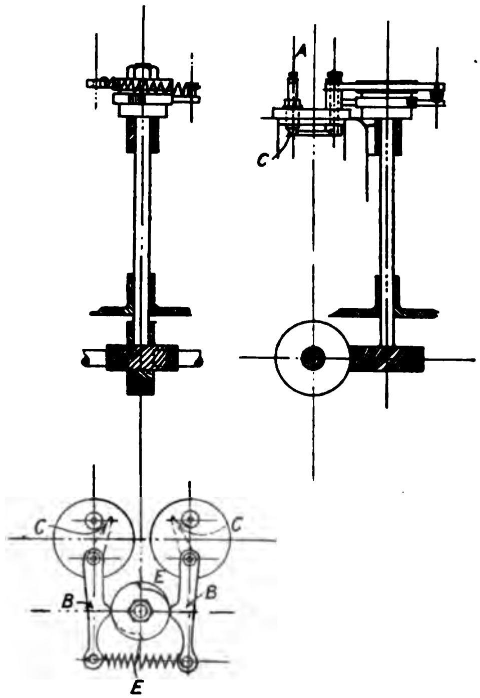


Fig. 59.

In this arrangement the cylinders are cast in pairs, so that each cam and cam shaft serves two cylinders with right- to left-hand plugs. Only one spring is used for each pair of cylinders.

Letters as before.

plug. The part through which the rocking arm passes can be insulated, and the other terminal be earthed; but this is not so convenient, as then all the striking gear has to be insulated. In order that the points should not wear away inside the cylinder the contact surface should be large. The gear may be arranged so that the moving finger strikes either the side or the end of the insulated plug, as indicated in fig. 60. It would seem best to make it strike the end, as it would then not be so liable to work the insulation loose, and it would be quite easy to make the end flat and of large diameter. The finger itself is often made very narrow, as in fig. 61, and consequently

burns away rapidly. Widening it, as in fig. 62, would improve it a good deal.

It may be mentioned that for some time most of the American launch engine ignitions have had the contact so arranged as to take place at the end of the plug.

Ignition Plugs.—The rocking arm which goes through the side of the cylinder may be arranged in several ways. The rocking arm with the insulated plug may be arranged on a separate plug, as in fig. 63, a very

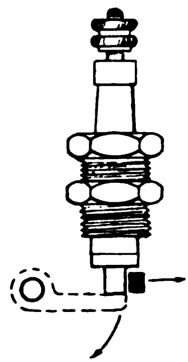


Fig. 60.

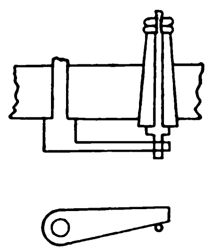


Fig. 61.

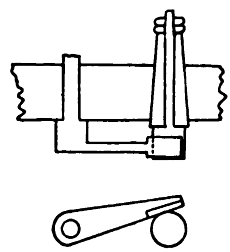


Fig. 62.

convenient way when the valves are at the side of the cylinders, as the whole plug caps over the valves; the ignition gear is then generally of the type of fig. 52, but as the size of the plugs necessitates the arm being rather short, it is difficult to get a quick break. Consequently, the larger the engine is the easier it is to put the plug in. Failing this, the usual arrangement is that shown in fig. 64.

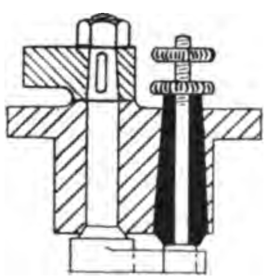


Fig. 63.

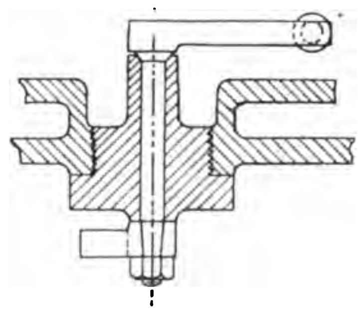


Fig. 64.

Sometimes the whole of the ignition is made in a small pocket at the side of the cylinder, as in fig. 65. This is a neat mechanical arrangement, but the joint required is awkward to make; theoretically, it would be better to put the ignition spark well in the combustion space.

The finger in the cylinder and the spindle that goes through the cylinder wall are generally made in one piece. The driving arm must be so fastened

that it will stand a good deal of hammering without becoming so loose that the ignition will not work. Sometimes it has been pinned on, which is not a satisfactory arrangement, though it will work if really well fitted. A more common plan is to fit it on with a cone and nut, as in fig. 66, but this requires a key that will prevent it from turning and ensure its being put on in the right place. Practically, the friction on the cone is no doubt a good enough fastening, but it is better that all parts should be so made to gauge that they will fit without adjustment.

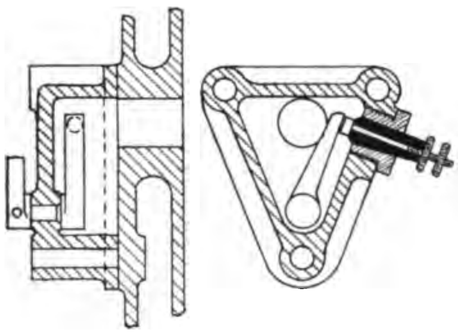


Fig. 65.

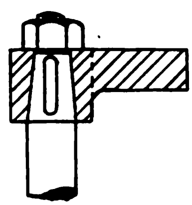


Fig. 66.

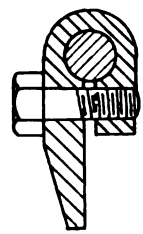


Fig. 67.

Another fastening that seems good, and is sometimes used, is represented in fig. 67. This is used on stationary engines which run continuously all day, and, therefore, have a great deal of wear, and seems very good.

The terminal passing through the cylinder is usually insulated, either by means of two soapstone washers, as in fig. 68, held in place by the nuts which screw up the terminal; or else by mica, arranged somewhat as in fig. 69—that is, by washers of mica pressed tightly.

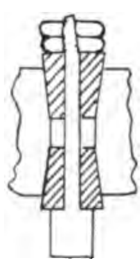


Fig. 68.

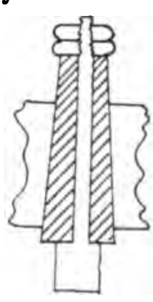


Fig. 69.

Timing of Low Tension.—The timing of low-tension ignition may be effected in various ways. By means of a worm and nut on the shaft working the make and break its relation to the crank shaft can be altered, but this involves the make and break gear being on a different shaft to the cams.

If the make and break is driven off a separate shaft by spiral gear, as in figs. 58 and 59, the time of ignition may be altered by moving either of these shafts longitudinally. Thus if, as in the figs., they are driven off the cam shaft, either the cam shaft can be moved longitudinally or the shafts work-

ing the make and break shifted vertically, either of which operation can be arranged for.

Another plan is to have spiral cams to the make and break, and to move the shaft carrying them longitudinally. This is not much used, and is not always easy to carry out, as, if the angle of advance is great relatively to the retarded position, the cams must either be very long or of a bad shape for wear, while it would be difficult to provide for the longitudinal travel.

The usual way of arranging the advance is to move the runner on the cam across the line of the shaft, as shown in figs. 52 and 57, which is most easily done when the trip rod has a rotating motion, as in figs. 53 and 57; all that is needed is to slide it vertically.

The make and break of low-tension ignition must be strongly constructed, the parts hammered on be made of case-hardened steel, and have an ample surface for the shock received. The arrangements must also be such that the timing of the ignition will not be upset by a small amount of wear. The runner on a vertically-pivoted rod, as in fig. 53, should bear on the full width of the cam, which it will do if the bevel is adapted to the varying angle of the runner, as in fig. 70.

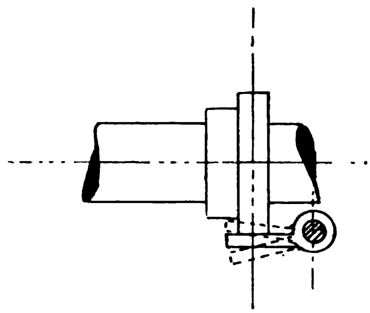


Fig. 70.

Vertically-sliding runners have much the same construction as those which work the cams (see Chapter vii.).

Comparison of Low- and High-tension Ignition System.—The low-tension system of ignition is evidently much simpler than the high-tension, and less liable to fail. With the high-tension system there may be (1) failure of the contact of the commutator; (2) of the trembler of the coil; (3) of the contact on the earthed circuit; (4) of the insulation of the high-tension leads; (5) or of the insulated plug in the cylinder; (6) or of some part of the low-tension circuit.

There have also been cases of the insulation of the secondary circuit of the coil failing, but this ought not to occur if the coil is properly made.

In the low tension the only likely difficulties will be the failure either (1) of the contact breaker, or (2) of the insulation of the wiring. The insulation of the low-tension circuit is not likely to give trouble in any case, but the liability is even less in the low-tension system than in the high, as the circuit is simpler. This will be seen on comparing the wiring of a low-tension ignition gear for a four-cylinder engine (fig. 71) with that of a four-cylinder with high tension (figs. 37 to 39). As the tension of the current in this is only at a very low voltage there is no difficulty. On the other hand, with the high tension there is an equally complicated low-tension system and all the high-tension leads in addition, and, as the coils are generally too bulky to be placed close to the engine, these are of some considerable length. As the tension is probably several thousand volts the least damp, or even oil, will cause the current to be short-circuited. The great objection urged against the low-tension system is that it has a moving part inside the cylinder. As all engines have pistons and valves moving inside the cylinder there is no reason why this should be considered a very great objection, and, in practice, there is no reason why the contact

breaker inside the cylinder should give any more trouble than the one outside it. The latter is always a very delicate piece of machinery, and often depends on extremely delicate springs and magnetic attractions. On the other hand, all the gear of the make and break in the low-tension system can be both much more substantial and independent of delicate springs.

It is also a great advantage of the low-tension system that if anything goes wrong it is pretty certain to be a mechanical fault in the contact breaker, whereas in the high tension it may be either this or an electric fault which may be difficult to find.

If the low-tension system is used, however, it *must* be well designed and substantially made.

The popularity of the high-tension ignition system is probably due to the ease with which it can be designed. The low tension has to be built into the engine, and, to be satisfactory, must be carefully designed. On the other hand, with the high-tension system the coils, accumulators, commu-

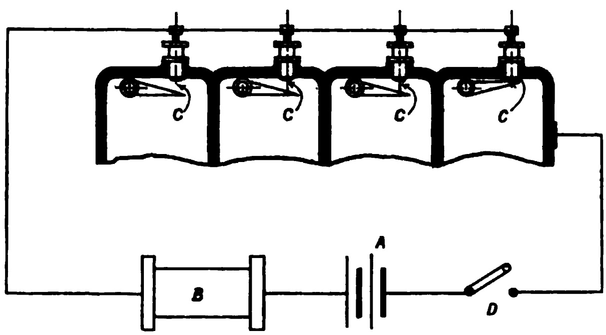


Fig. 71.

A, Battery.
B, Coil.
C, C, C, C, Contact points.
D, Switch to cut off current when engine is not running.

If a magneto is used it takes the place of both the battery, A, and coil, B, and the switch, D, is often omitted.

tator, &c., can be bought, and the only designing is fixing a commutator on the engine and making holes in the cylinder for the ignition plugs. This will not have any weight with the makers of high-class engines and cars, but it is noticeable that the low-tension system predominates in the best cars, and the high-tension one in those of inferior make.

Magneto Machine.—This has latterly become the usual source of electricity, mainly owing to the influence of Messrs. Sims & Bosch, who introduced a special form of magneto for the purpose, but other forms are used.

With the low-tension system the most natural way of doing away with a battery is to use an ordinary dynamo. This has been done with stationary engines a great deal, and in this case all that is necessary is to drive a small dynamo from the engine with a belt or other means. The only difficulty which arises is that if the dynamo gives sufficient current to start when the engine is turned by hand, it will give an excess when it is running at full speed. This effect will be greatly intensified, since the current produced

increases far more rapidly than the speed of the armature rises. This can, to a certain extent, be remedied by fitting the dynamo with a centrifugal governor so arranged that at a certain speed it will throw out a clutch and prevent any further increase of speed. Or the engine may be started by means of a battery or accumulator, and then run on the dynamo; but this is rather complicated.

If, for a dynamo which has electro magnets, a magneto with permanent magnets is substituted, the matter is much more simple, as a well-made magneto can be used for all speeds. There is the further advantage that the self-induction of the armature of the magneto adds to the spark, and therefore the self-induction coil may be dispensed with. This is not the case in the dynamo, as the current from the armature can go across the shunt through the field magnets. In the case of the magneto, the current will not increase very much with an increase of speed beyond a certain point, as the circuit is closed for a shorter time and the self-inductions will prevent too great a current flowing.

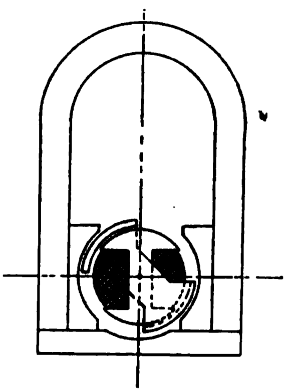


Fig. 72.

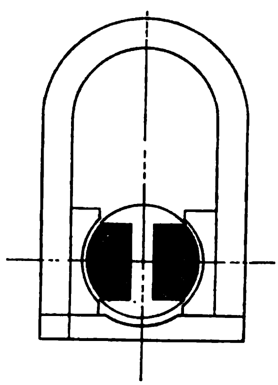


Fig. 73.

The armatures for these machines may be the ordinary ring or drum armatures; but, in Europe, an alternating machine which keeps step with the engine has generally been adopted. A simple shuttle-wound armature and no commutator will give an alternating current, and a good spark is obtained by arranging that the current is at its height when the break occurs. In this case the magneto must be driven by gearing at a speed bearing some proportion to that of the engine. This is the usual form of magneto ignition on cars.

The principle of the special form of magneto introduced by Messrs. Sims & Bosch is shown in fig. 72. It has stationary armature and field magnets, and a shield which alternately causes the lines of force to flow through the coils of armature and round the ends beyond the coils. This shield may be either rocked, as in the early form, or revolved, as is more often the case now. With a revolving shield the current reaches a maximum four times in a revolution, so that for a four-cylindered engine the machine need only go round at half the speed of the engine to obtain a spark for each cylinder. The absence of brushes is an advantage.

Most car-makers, however, use simply an alternating magneto with an

ordinary H-section armature. A section of such a machine is shown in fig. 73. In this the armature can be of larger size than in the shield type, as there is more room for it, and should, therefore, give a larger spark for the same sized machine, more especially as there is only one air gap instead of two in the magnetic circuit. Messrs. Sims & Bosch now make a machine of this type as well as the revolving shield type. There must be a brush to collect the current from one part of the armature wire as the other end is earthed, but this does not give any trouble in practice. A magneto can be made with the armature fixed and the field magnets revolving. Theoretically, it is the right plan, and a few firms have adopted it by making the magnets part of the flywheel. This allows of the magneto being much lighter as the field magnets form part of the flywheel. There is no reason why a magneto with revolving magnets should not be made separately, but no one has done this.

In magneto machines with alternations keeping step with the engine, the magneto must be capable of working over a great range of speed, as the engine can only be run at a slow speed to start it, and there must be sufficient spark for this. It is usual, therefore, to wind the armature of machines of this type with a great deal of rather fine wire. The diameter of the wire is about 30 legal standard wire gauge, and the length varies from about 150 yards in small magnetos for cycles to over 350 in larger ones for cars. As compared with the battery and coil arrangement, the magneto works with a smaller current, but with a higher voltage and a much longer circuit for the self-induction. The battery and coil can easily be made to give a larger spark than I have ever seen from a magneto. One practical point is that the insulation needs to be more perfect with the magneto than with the battery and coil, presumably on account of the higher voltage. I once had a case of an engine with low-tension ignition which would run perfectly with a coil and battery, but, if the magneto was used, the insulation in the plug of the cylinder broke down.

For small engines, such as are used in cars, the magneto is so much the most convenient that it will probably become universal, but, for larger stationary and marine engines, the battery and coil has many advantages and may stay in use.

Many arrangements have been made by which the reversal of the magnetic lines is made more sudden, and a better spark thereby obtained at low speeds. One plan is to have the magneto itself worked by a cam, so as to get a very sudden movement. This is used satisfactorily on stationary engines in which the speed is slow, but does not answer for high speeds. The same result is obtainable by so shaping the pole pieces as to get a very sudden change, or by running the magneto a good deal faster than the engine. Any means by which this may be done gives a better spark at low speeds, provided the break comes at exactly the best point for the magneto, but reduces the range over which it will work. That is to say, it makes it more important that the break should be exactly in step with the magneto. It is very common to have the ignition point adjustable to a certain extent without altering the relation of the magneto to the engine, and, if the magneto is of ordinary construction, this can be perfectly well arranged, and a good enough spark obtained at all points for all practical purposes.

It is very doubtful if there need be any adjustment to the ignition point with a magneto, provided the arrangement of the break is such that the spark really does take place at the same point of the stroke at all speeds. The

"advance" of many ignitions does not really advance the spark, but rather takes up the lag of some spring. With a magneto, in starting, the engine must have considerable speed to yield a spark, and, in this case, the engine will not "fire back," even though the spark takes place some time before the end of the stroke. On the other hand, as the engine goes faster the spark gets bigger, which, to a great extent, has the same effect as advancing it.

Magnetos have also been fitted to a high-tension system. This may be done in several ways. As in the low-tension, a dynamo may be used for the running and a battery for starting, and this can be made to work very well; but an alternator keeping step with the engine is the arrangement generally used.

Every variety of the high-tension system with accumulators may be used with the magneto. Probably the most usual just now is to have a single-contact spark with one induction coil and a high-tension distributor. In some cases, the high- as well as the low-tension coil is on the armature of the magneto, and in this case is very convenient, as all the leads are self-contained and there are no outside connections. Sometimes the coil is mounted in the top part of the magnets, which is also very neat. Sometimes the coils are separate, and then the leads are about as long as those in the ordinary high tension with accumulators. If one coil only is used, wound on the core of the armature or separate, there must be a high-tension distributor.

The construction of a high-tension magneto is a speciality of several firms who make ignition apparatus, and varies very much with each maker. The reader is, therefore, referred to their catalogues for information respecting these.

Whether the high-tension magneto will supersede the low-tension or not is at present impossible to say; for, although the accumulator is done away with, most of the other difficulties apparently remain. It is true that, when the single coil used forms part of the magneto, a great many of the outside leads are suppressed, while the only leads apparent are those from the machine to the respective sparking plugs. Still, the connections are all there, including the high-tension distributor, and putting them all in one case only complicates matters should anything go wrong, and all the difficulties of insulation remain. Many of the makers of high-class cars have faith in it, but a still larger number of makers of the best class believe in low tension, good results being obtained from both systems. On the other hand, as previously mentioned in discussing the question of accumulator ignition, it is obvious that a good many of the smaller makers seem to have taken up the high-tension magneto because it does not entail any designing. They buy the magneto complete, and can generally stick it on their engine somewhere or other and drive it somehow. That the French should have a weakness for high-tension ignition is natural, seeing that it is the original arrangement they used both on gas engines and cars; but an inspection of many of the smaller makes of cars gives one the impression that the makers have adopted the "magneto" ignition for the high-tension system, regardless of the fact that its reputation was based on experience in connection with the low-tension apparatus.

Double Ignition.—Cars are often fitted with two complete ignitions. This has many advantages. Probably the best combination is low-tension magneto and high-tension with trembling coils, as the two systems are quite independent, and the latter makes it easy to start the engine; while the use

of both develops the maximum power of the engine. Other combinations can be used, but the greatest advantage is obtained when the two ignitions are absolutely separate. In high-tension arrangements with the two systems, each should be complete with its own coils, plugs, &c.; for otherwise, if anything goes wrong, the fault may be in a part common to both, when the inconvenience is just as great as with only one ignition. A combination sometimes used is to have a low-tension magneto and a "supplementary" ignition with low-tension coil and an accumulator working on the same contact breaker. This is convenient for starting and is very cheap to fit, as the low-tension coil costs about 10s. retail and accumulators are not costly. It will also work, even if the magneto itself breaks down, but there ought to be no excuse whatever for this. Magnetos only work at a very low voltage compared with coils, and can, therefore, be made proportionately free from breakdown.

CHAPTER V.

CARBURETTORS.

Definition.—The object of the carburettor is to make an explosive mixture of petrol and air suitable for use in the engine cylinders. Other fuels are sometimes used, and will be briefly noticed in a future chapter; but the usual fuel is petrol, which is petroleum spirit having a specific gravity of from ·680 to ·720, and completely volatile in air without the aid of artificial heat, though in many cases it is advisable to warm the carburettor slightly, as the petrol may be cooled too much by the rapid evaporation.

The exact proportion of petrol to air which is the best to use in a petrol engine probably depends a good deal on the requirements. It is known from gas engine practice that the mixture which gives the greatest power for the smallest cylinder is not that which gives the greatest power for the least gas. It is, therefore, usual in gas engines to use a mixture a good deal weaker than that which gives the greatest power for the smallest cylinder, in order to save gas. At present, in the petrol engine, it is far more important to get the best power out of the engine than to economise the fuel, and the preference is for the mixture most suitable for this purpose.

Another limitation to the use of a poor mixture is that mixtures of petrol vapour and air are not inflammable over such a large range as those of gas and air.

It is not quite clear if the same mixture is required under all different conditions of engine running, or if it should vary as the engine is run fast or slow, full open or throttled, but probably no very great variation is required. Theoretically, there ought to be some variation at all events when the engine is throttled, as in this case the proportions of burnt gases in the charge is much greater than when running full open. The right proportion of liquid petrol to air for perfect combustion in theory is about 1 volume to 8,000, and experiments are in accordance with this. The maximum amount of vapour that air will take up at ordinary temperatures is over ten times this, so that it will take up far more than there is oxygen available for combining with it. Thus, air saturated with petrol vapour at ordinary temperature is *not* an explosive mixture, but must be diluted with more air in order to become explosive, a property taken advantage of in making the surface carburettor, which is the simplest form.

Surface Carburettor.—This consists of a tank for the petrol, an inlet and outlet pipe for the air, and baffles for making the air travel over the surface of the petrol until it is saturated with vapour. This is mixed with more air and used in the engine, much as coal gas is used in a gas engine, and the proportion of air that has to be added is about that required for coal gas. In motors the gas and air are usually mixed by being passed through a three-way cock, regulated by hand, according to the mixture desired. There are variations in this arrangement, such as wicks for absorbing the petrol and giving more surface to the air. In many the vaporiser is a small

vessel continually fed from a larger tank. In all cases the vaporiser requires to be warmed in order to counteract the cooling effect of evaporation.

The great objection to this class of carburettor for automobile work is that the mixture which comes from it is liable to continual variation. The amount of petrol which air will take up varies with the temperature and moisture, and also with the density of the petrol. Where the tank and vaporiser are combined in one the more volatile parts of the petrol are taken first, and, therefore, the proportion of petrol the air will take up gets less and less, and the mixture valve has to be constantly altered to get a good mixture. Where there is no governor this is not a very serious matter, as the valve can be continually altered; but where there is a governor it is not always evident whether the mixture is right, and, therefore, the engine may be run for a long time with much too rich a mixture without the driver being aware of it, and this is both wasteful of petrol and productive of much smoke.

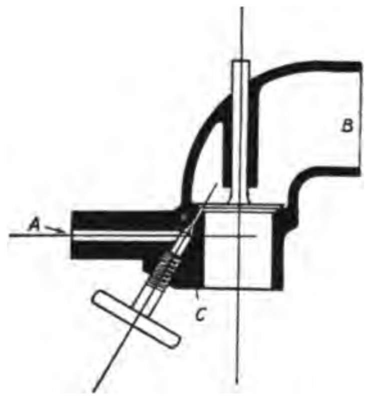


Fig. 74.

A, Petrol inlet to carburettor.
B, Inlet pipe to engine.
C, Needle valve to regulate flow of petrol.

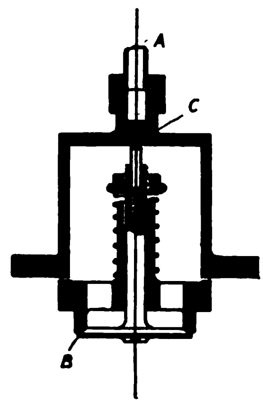


Fig. 75.

A, Petrol pipe.
B, Inlet valve to cylinder.
C, Petrol valve, which is opened by the valve B when latter opens.

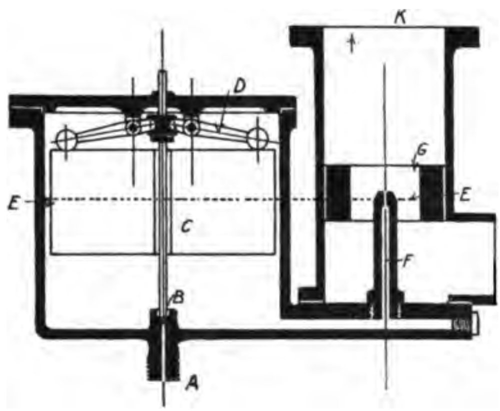
There are several great advantages in the surface carburettor, the main one being that it completely evaporates all the petrol, while none of it goes into the cylinder in the form of spray. This should make for economy. There are also no delicate parts to get out of order. Its action is not stopped by dirt or water. Nevertheless it has very nearly gone out of use, though a few firms still employ it, and have obtained good results.

The next form of feed to notice is the mechanical pump. No doubt the most perfect feed would be a pump to deliver at each stroke of the engine the right amount of petrol for it to use. This is difficult in practice in a motor, as the endeavour is to reduce the power by throttling, and not cutting-out explosions, to do which the air and petrol must be reduced in proportion, which is not easy to do. It is, however, used on stationary engines and on one or two marine ones with cut-out governors, but is not used on automobiles.

Another type is that used in the United States for a great number of

years. In this a valve placed in the suction pipe is sucked open by the engine, and this is followed by the opening of the petrol valve. Fig. 74 shows a very simple arrangement. In this the petrol supply comes into the seating of the air valve, and the two passages are opened together. Occasionally the petrol valve is a separate valve placed on the end of the inlet valve of the motor, which must be automatic. Fig. 75 shows this arrangement, which can also be used as a separate carburettor.

In this type the flow of the petrol is induced partly by the vacuum caused by the engine sucking, partly by the pressure from the head of petrol in the tank. The latter decreases as the petrol is used up. In practice, also, the petrol has to be carefully regulated for different speeds of the engine in order to give a satisfactory mixture. It has been long and extensively used in the United States for launch work, so that almost every possible combination has been tried. It answers well for a launch engine in which the revolutions are more constant than in car engines, and the ignition generally used is a low-tension one, which gives a good spark that will ignite mixtures of various strengths. A similar arrangement suits



A, Petrol inlet.
B, Needle valve.
C, Float, which closes the needle valve, B, through the levers, D, when the petrol reaches the level, E, E.
F, Petrol jet.
G, Air nozzle.
K, Inlet pipe to engine.

Fig. 76.

stationary work with heavy oil, but in this case the engines run at quite a constant speed with a cut-out governor.

Float Feed Carburettor.—The most generally used type of carburettor is the float feed, the simplest of which is indicated in fig. 76. In this the petrol is kept at a constant level in a chamber by a float and valve, and, connected with it, there is a petrol jet in an air nozzle. The petrol is kept just about the top of the jet. When the engine sucks, a vacuum is formed in the air nozzle, and both petrol and air flow into it. The petrol comes out of the jet in the form of spray, and is carried up the inlet pipe and vaporised there. According to the ordinary laws of the flow of gases and liquids, both the petrol and air flowing into the nozzle ought to be proportionate to the square root of the vacuum. This being so the proportion of petrol to air ought to be constant at whatever rate the air was sucked through. As a matter of fact it is not so if the carburettor is worked over a great range of speed of flow, and some special means have to be taken to keep the mixture at all constant, which will be referred to later.

Taking the float first, there are four ways of making this, all of which are in use by different makers. In the first of these, the float as it is raised depresses the needle valve by means of levers hinged to the cover of the float chamber, as in fig. 76. This is a very common form, but has the disadvantage that it is not easy to arrange it so that the cover can be taken off without taking out the needle valve. This prevents one seeing easily whether the valve is leaking. The valve is lifted by the weight of the end of the levers, and there are usually two collars between which the levers work to raise and lower it. It is not quite easy to get these to work freely and without ever sticking and, at the same time, not to have too much slack.

The next arrangement is that shown in fig. 77. This is the same as the last inverted, but the valve is closed entirely by its own weight, and

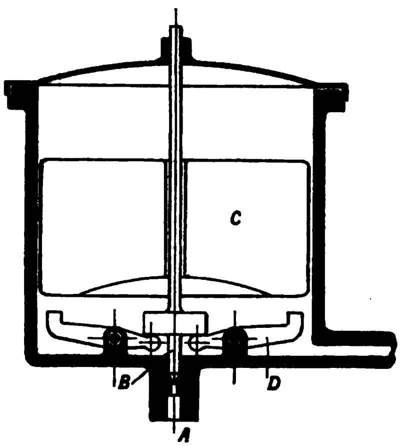


Fig. 77.

A, Petrol inlet. C, Float.
B, Needle valve. D, Levers.

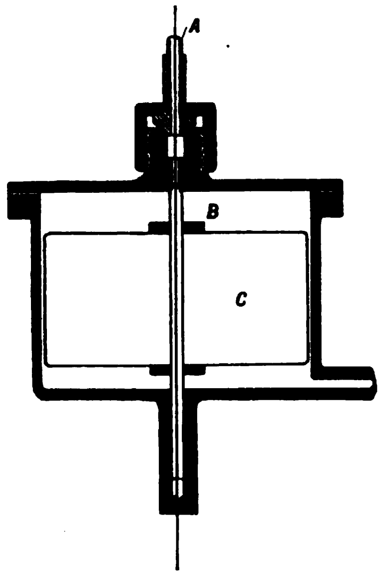


Fig. 78.

A, Petrol inlet. B, Needle valve.
C, Float.

only opened by the levers when the weight of the float comes on them. According to my experience this is much less liable to stick than the other. It is also more convenient to take to pieces, as the float can be taken out without interfering with anything else.

In both these the valve usually goes through a hole in the float formed by a tube soldered in. This is a cheap arrangement to make, as the whole can be machined in one centre, but it makes another joint in the float to leak, and sometimes the valve is put at the side. In this case and in the next types there is no reason why the float should not be spun so as not to have any joint below the level of the petrol. This would be an advantage, as floats sometimes do leak and then the petrol overflows. When the car is in action this is generally discovered by the bad running of the engine, but if

it happens when the car is standing the petrol may overflow and cause a bad accident.

A simpler kind of float is that shown in fig. 78, in which the valve is soldered direct to the top of the float. This is the cheapest of all as there are no levers to fit, but it is only accessible by disconnecting the petrol pipe. This objection is avoided in the form shown in fig. 79, but it is difficult of access when it is necessary to grind in the valve, otherwise it seems good.

Float chambers should always be provided with a space for the settlement of dirt and water. The simplest way of arranging this is to make the passage to the jet some way up in the float chamber. If this is not done, there should be a settling place in the pipe just below the carburettor, with a cock in it, so that when a little petrol is drawn off the dirt and water issue with it.

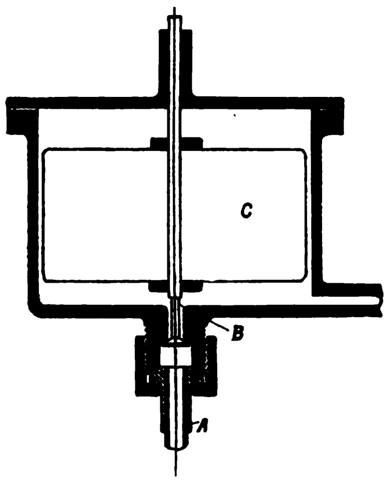


Fig. 79.

A, Petrol inlet. B, Needle valve. C, Float.

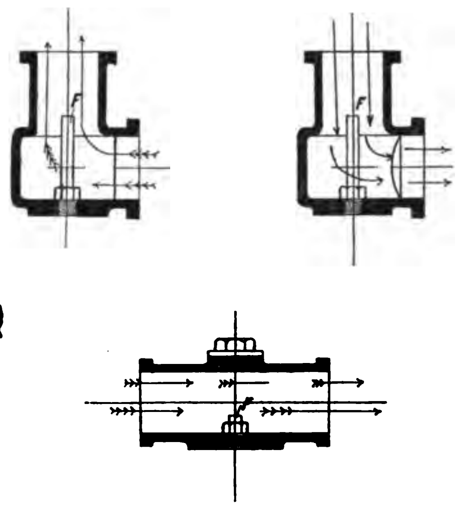


Fig. 80.

F, Petrol jet in each case.

The direction of the flow of air with regard to the jet may be various (see fig. 80). It may be as shown in fig. 76, or it may be reversed—*i.e.*, the air may flow in through what is there marked as the outlet. It may also flow across the nozzle. Any of these plans suit if the carburettor is properly proportioned. As mentioned above, the carburettor, in practice in its simplest form, will not give a constant mixture, the tendency being for the mixture to become richer as the rate of flow is increased. Various reasons have been suggested for this.

The range of flow over which carburettors are expected to work is now much greater than in former times. Engines are now expected to work over a range of speed of something like 10 to 1. That is to say, if the maximum is 1,500 revolutions the engine is expected to work smoothly down to 150. As it may, at this low speed, be also very much throttled, the range over which the carburettor has to work is quite 30 to 1. This

is an enormously greater range than the earlier carburettors were required to work over, and special means must be taken to secure a good mixture throughout it.

The tendency of the mixture is to get weaker as the rate of flow decreases, and it will be evident, on looking at the construction of a carburettor, that there will be a minimum flow below which no petrol will be sucked through the jet. The object, then, is to devise a plan for increasing the airway in proportion to the increased flow through the carburettor. The mixing valve of the type shown in figs. 74 and 75 would seem to meet the case as it gives a constant head of petrol, but it has not come into general use owing to its irregular action.

Regulating the mixture may be effected in various ways, either by hand or automatically.

The regulation by hand-moved arrangements comprise the following means of correction :—

1. The jet and air nozzle may be arranged for giving too rich a mixture, and an additional valve, controlled by hand, be used to supply the air required for a mixture of the right proportions. This is a common plan, and, on the whole, works well, but seems to require a good deal of alteration for different engine speeds, &c.

2. The size of the petrol jet may be varied by means of a screw-down valve, but, being delicate to work, it is not much used.

3. The size of the inlet to the carburettor may be varied by means of an adjustable valve to the inlet, or by sliding the air nozzle up or down. This works very well, and, generally speaking, does not require such constant adjustment as plan 1.

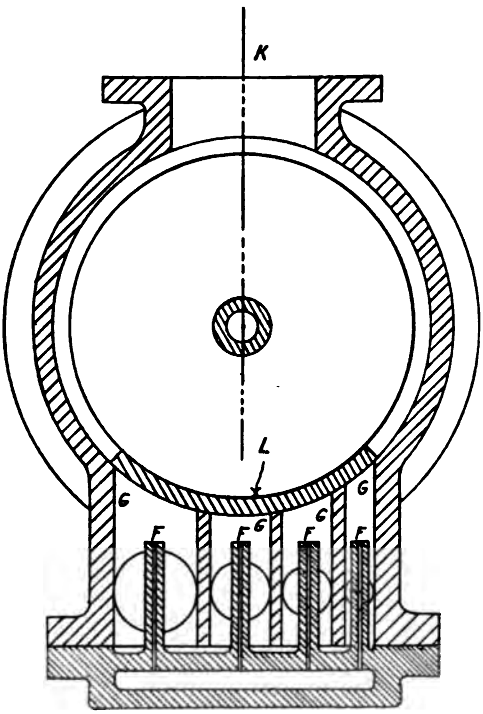
4. A greater or smaller number of nozzles may be brought into play. Fig. 81 shows an arrangement which does this, the throttle and carburettor being in one piece.

The objection to this, and all arrangements controlling the mixture from the throttle, is that it takes no account of the engine speed. That is to say, the carburettor is the same when full open whether the engine is running fast or slow, although the amount of flow through the carburettor is, perhaps, ten times as much in one case as the other.

It will be seen from fig. 81 that a succession of carburettors are fed from one float feed. These may be opened either in succession simultaneously or, singly, one after the other, and the smaller ones behind closed. This is rather expensive, as the correct adjustment of four carburettors has to be provided for; but the range is almost unlimited.

The preference is for controlling these adjustments automatically. In the plan last noticed, the adjustment is connected with the throttle; the extra air supply might be made to depend on the throttle by varying the size of the inlet or petrol jet. This may be better than having a carburettor devoid of an automatic arrangement, but it is not perfect, as it takes no account of the speed of the engine. Thus, if an increase of speed is desired when the engine is running slow and throttled, the mixture supplied must be rich; but, if the throttle is opened full and the air-supply inlet is also opened full, the engine will not get the rich mixture required. If the throttle is hand-controlled, this defect may be met by not opening the throttle full until the speed has improved, but to do this properly requires very careful driving. If the throttle is worked by a governor, the proper mixture cannot be ensured. If the air supply is controlled

by the speed of the engine, a more regular mixture will be obtained as the flow through the carburettor varies much less with the throttling than the engine speed. The internal friction of the engine is, probably, seldom very much less than 25 per cent., while the consumption per horse-power will be slightly greater when throttled; so that, at a given speed, the consumption when fully throttled will be about a fourth of what it is when full open. On the other hand, as mentioned, the speed of the engine may vary in the proportion of 1 to 10 or more. Further, if the adjustment is correctly made for the engine speed, the engine will always get a good mixture when the throttle is full open at any speed, even if it



F, F, F, F, Petrol jets in
G, G, G, G, Nozzles.
L, The throttle, opens large nozzles successively to give the engine more mixture.
K, Inlet to engine.

Fig. 81.

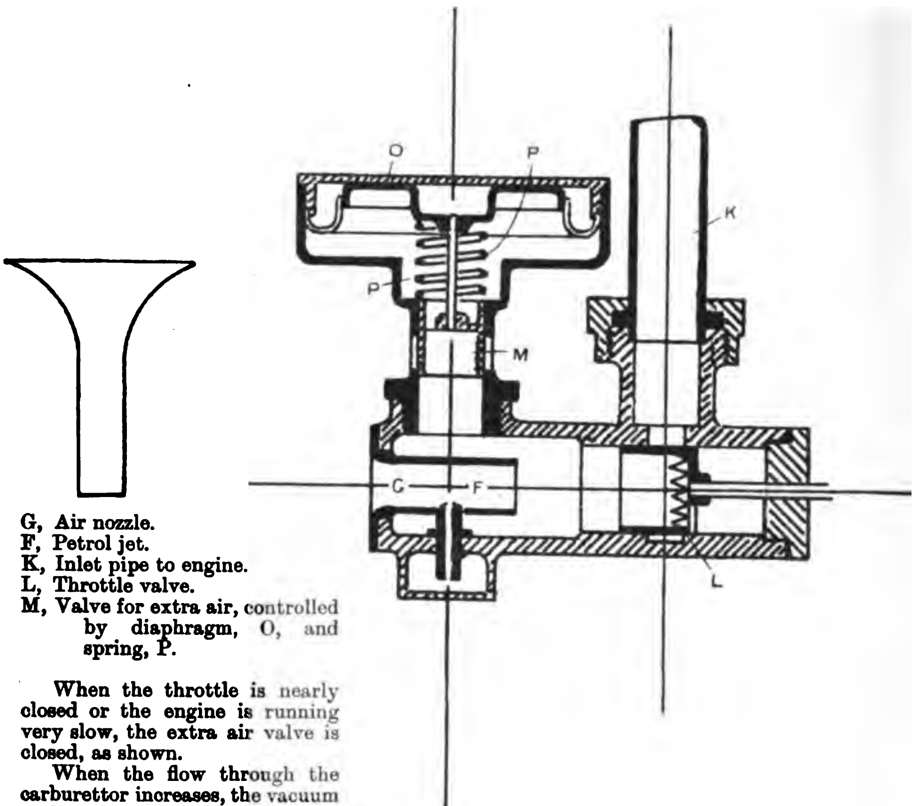
does not do so when throttled. In this case, if the throttle is worked by a governor, should the engine be throttled and the mixture fail to be good enough to explode, the engine will immediately slow down and the throttle will open. Then the mixture will be right and the engine will not stop, as it would do if opening the throttle gave it more air.

Several different means have been used to effect the control of the mixture from the engine speed, such as a centrifugal governor or a diaphragm worked by the pressure of the circulating water. In either case, they usually open an extra air supply like the undermentioned plans.

Carburettors thus made cannot supply a constant mixture, as no account is taken of the engine speed by the one and of throttling by the other. A

carburettor controlled by the vacuum in the suction pipe itself would, theoretically, give a constant mixture under all the alterations in the rate of flow produced by throttling or by the engine running slow, and be suitable for engines of all sizes, whatever amount of gas is required. This is unattainable in practice, but a carburettor of this type ought to supply a mixture constant over the range necessary for one engine.

Automatic Carburettors Controlled by Vacuum.—In most of these the nozzle and jet supplies give a mixture of the right proportion when the minimum quantity required passes through the carburettor; the further



G, Air nozzle.
F, Petrol jet.
K, Inlet pipe to engine.
L, Throttle valve.
M, Valve for extra air, controlled by diaphragm, O, and spring, P.

When the throttle is nearly closed or the engine is running very slow, the extra air valve is closed, as shown.

When the flow through the carburettor increases, the vacuum sucks the diaphragm, O, down and opens the valve, M.

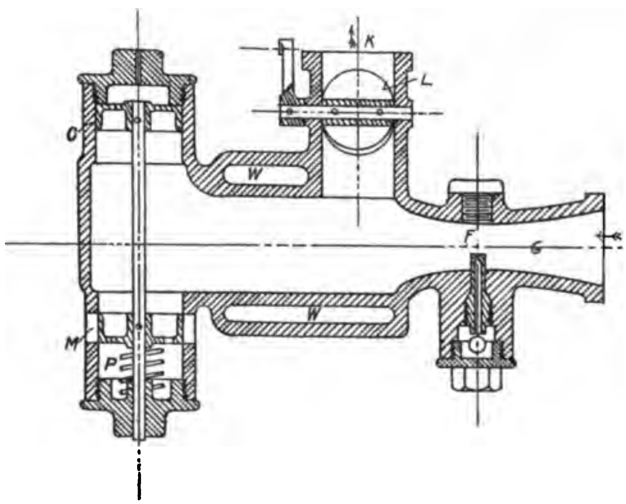
Fig. 82.

addition of air needed being regulated by the increased engine speed or the throttle being opened. A very early form of it consisted in simply putting an air valve controlled by a spring on the suction pipe near the engine, the valve being exactly like an ordinary automatic inlet valve. This was suggested six years ago, but did not come into use, probably because engines were not then run over a great range of speed and generally had cut-out governors.

The first which came into general use is the Krebs, the principle of which is shown in fig. 82. In this the extra air valve is a piston valve

controlled by a diaphragm, held in place by a spring. This diaphragm, with the piston valve and the rod which works it, is necessarily heavy, but the motion is steadied by the hole at the back of the casing to the diaphragm being made so small that the air can only get in and out slowly. This carburettor, as made by Messrs. Panhard & Levassor, undoubtedly works extremely well.

The same principle may be carried out with a piston instead of a diaphragm, as shown in fig. 83. In this, however, we have a disturbing element not met with in the diaphragm—viz., the leakage of the piston. As mentioned, the hole at the back of the diaphragm must be a small one, or the rapid motion of the piston would destroy the mechanism. No difficulty arises if there is no leakage; but if there is a slight leakage, as in the case of the piston, the hole must be of the right size to ensure the necessary vacuum. The size will vary with the constant wear of the piston and the consequent variation in the leakage past the piston. Further, the



F, Petrol jet.
G, Air nozzle.
K, Inlet pipe to engine.
L, Throttle.
M, Extra air valve.
O, Piston controlling this.
P, Spring.
W, Hot-water jacket.

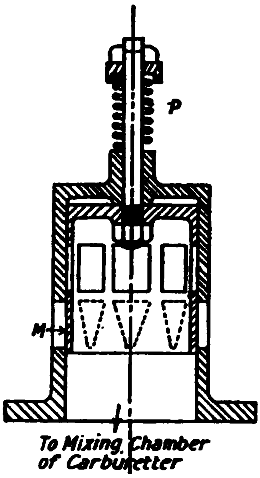
Fig. 83.

piston will be more liable than the diaphragm to be stopped working by dirt. The small hole has the effect of retarding the action of the valve in supplying air to the engine; if it is very small (as it must be if the parts are heavy), the valve will open so slowly when the throttle is opened suddenly that several charges of very rich mixture will enter the engine before the valve opens, thereby checking the action of the engine at the very time it is wished to quicken the pace. A simpler form of the piston arrangement is shown in fig. 84, which can be made very light, as the piston and valve are in one piece. It has the additional advantage of being very cheap.

In these valves the shape of the slots admitting the air is peculiar, being somewhat as in fig. 82, in order to supply a large addition of air for the first part of the movement of the valve, and a much smaller one for the same amount of movement afterwards. The shape of the slots will obviously depend entirely on the rate of increase in the strength of the

spring, and, therefore, the slots and the spring will have to be made to suit each other.

At present many makers are in favour of using the older plan of a valve, like an ordinary inlet valve, for the extra air supply. It has several advantages, such as lightness (so that there is no need for a dashpot) and cheapness. Old non-automatic carburettors can often be considerably improved by placing a valve of this kind anywhere between the carburettor and the engine. Valves are, in fact, on the market suitable for being attached to the inlet pipe, and are made to fit different sized inlet pipes.



M, Combined valve and piston, which is sucked down by vacuum against the spring, P.

Fig. 84.

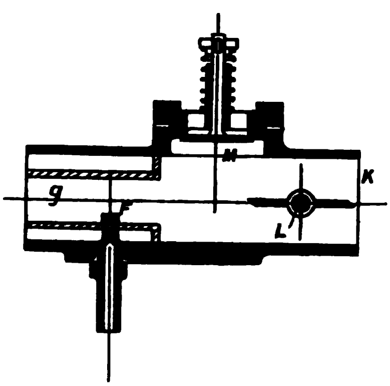


Fig. 85.

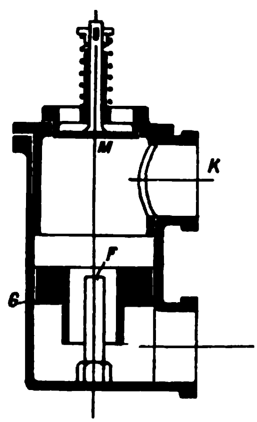


Fig. 86.

F, Petrol jet.
G, Air nozzle.
L, Throttle valve.
M, Air valve.
K, Inlet to engine.

The best plan is to make the automatic valve part of the carburettor, as

it is cheaper to construct, and also allows of a more thorough mixture of the air and petrol vapour before passing through the inlet valve. Figs. 85, 86,

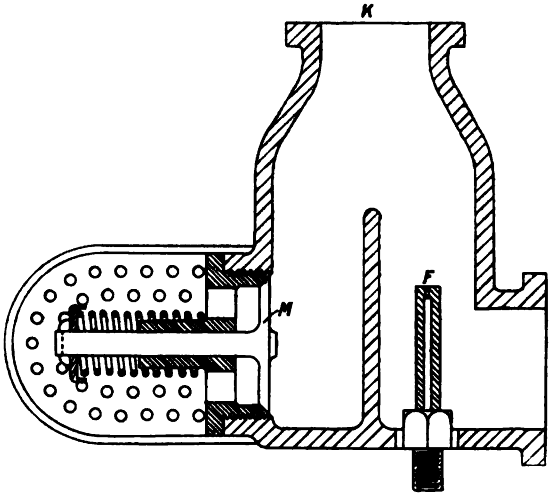


Fig. 87.

F, Petrol jet. M, Air valve. K, Inlet to engine.

and 87 show three forms; but there are many others of a similar nature. Fig. 86 is a very convenient arrangement to make, as the extra air valve, the jet, and the throttle valve are all on the same centre.

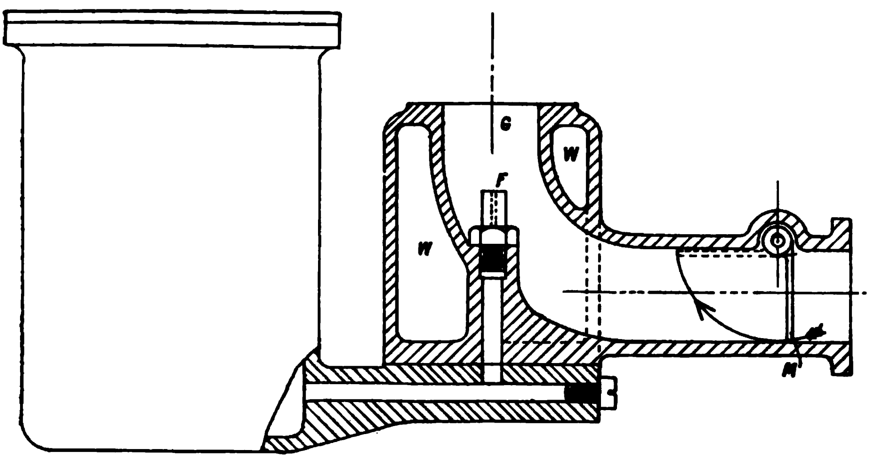


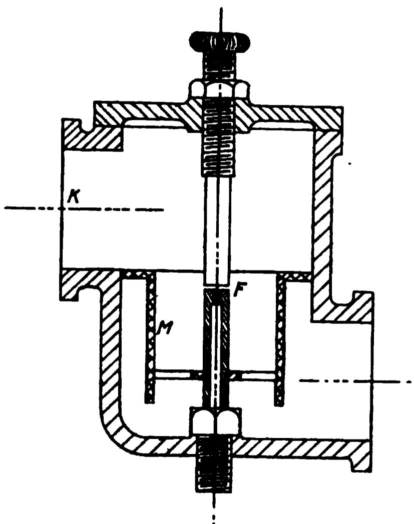
Fig. 88.

M, Valve, automatically increasing vacuum at low speeds.
W, Hot-water jacket.

With the simple air valve of the inlet type there is no graduation of the passages, except that given by the spring. There is no reason why a spring

should not be arranged to give the same graduation of opening for given increases of vacuum, as in the case of the piston valve. It is simply a question of making a spring that will give the right lift with a given vacuum, and if one spring is not sufficient, a second could be added, to come into action when the first was compressed to a certain amount. Or the opening could be graduated by tapering the passage of the valve, so that a full opening would need a high lift, and the shape of this opening be made as desired.

Theoretically, better results should be obtained by automatically varying the size of the air inlet to the carburettor, but it has not been so much adopted as the last plan. In the arrangement shown in fig. 88 a valve in the inlet obstructs the flow of air, but opens wider under increased suction. An ordinary inlet valve or a piston valve of the type previously described may also be used, but the spring controlling it has to be so light to get the flow right at different speeds that the valve is liable to stick. Another plan is to have a lifting nozzle, as shown in fig. 89.



M, Air nozzle and automatic valve combined.

When the rate of flow through the carburettor becomes slow, M falls by gravity and reduces the area of the passage below it.

Fig. 89.

In these the difficulty seems to be that, although the vacuum will suck the right amount of petrol out of the jet at low rates of flow, there is an insufficient draft through the carburettor to evaporate it properly. Consequently, liquid petrol is liable to accumulate in the carburettor, and when the throttle is opened suddenly the engine sucks it all up, and the mixture contains too much petrol. There are several remedies for this, such as contracting the air passage at the petrol jet, as in one of the oldest forms of automatic carburettor, which is shown in fig. 90, but did not then come into general use, probably because there was little need for it, when the range of the engines was smaller than it is now, and it might have required a dashpot.

The same principle may be applied to the lifting flap, as this may be put right over the petrol jet.

Other possible improvements are hot-water jackets for the parts where the petrol lodges, and making the petrol flow continuously down hill to the inlet valves.

As aforesaid, it is more usual to add air to a rich mixture than to adjust the mixture in the way just described, but if the above difficulties can be overcome this type may be more used in the future.

Varying the petrol jet and air nozzle together, as shown in fig. 91, should give a constant mixture over a practically unlimited range; but the difficulty is to make a satisfactorily efficient needle valve in the petrol jet. This type is used to a small extent only.

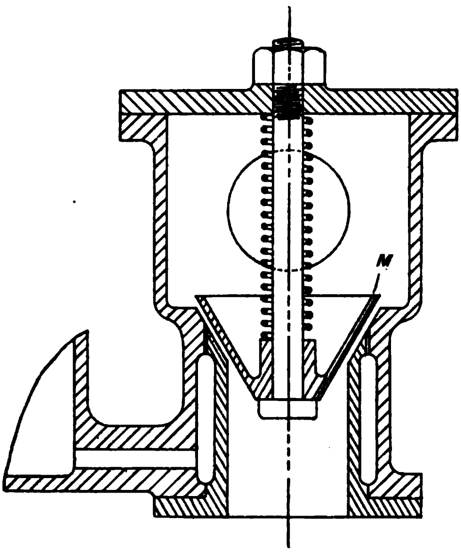


Fig. 90.

As the engine sucks harder the valve, M, rises and enlarges the air nozzle.

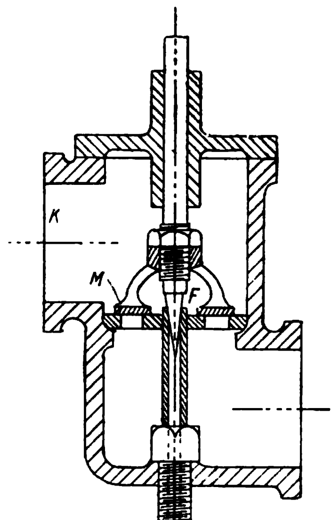


Fig. 91.

F, Petrol jet.
M, Valve, opening petrol jet and air nozzle in proper proportions.
K, Inlet to engine.

A succession of carburettors can be worked automatically as well as by hand, and should give a very great range indeed. Fig. 92 is a diagrammatic view of an arrangement of this kind with three nozzles. This should give a very great range indeed. It may be assumed that the ordinary single nozzle and jet will work satisfactorily over a range of flow of about 4 to 1, and give a mixture which is sufficiently uniform for all practical purposes. Hence each nozzle and jet is made four times the size of the one before. If the minimum rate of flow through the carburettor is taken as unity, the range of the smallest nozzle will be from 1 to 4; then the second nozzle opens and has a range from 1 to 16; so that the range of the two together will be from 1 to 20; then the third nozzle opens,

and as its range is from 1 to 64, the range of the three nozzles will be from 1 to 84. The mixture will not be absolutely uniform, as the supply from each nozzle gradually becomes richer until the next nozzle opens, when it will be diluted with a rather too pure mixture coming from it. This will again get richer until the next opens; but the total variation of the mixture should not in theory vary more than that of one nozzle worked over a range of 1 to 4, which is probably quite as uniform as the mixtures supplied by most automatic carburettors.

If each nozzle has a range of 1 to 3, the total range will be from 1 to 39.

This principle would, therefore, seem to be well worth considering, as there is no difficulty in keeping up the rate of flow through the nozzle while retaining the full area necessary to give a full charge at full flow, but the increased complication and expense, together with the difficulty of making the parts light enough to work without a dashpot, may be objections.

All the above principles are used with success by different makers, so

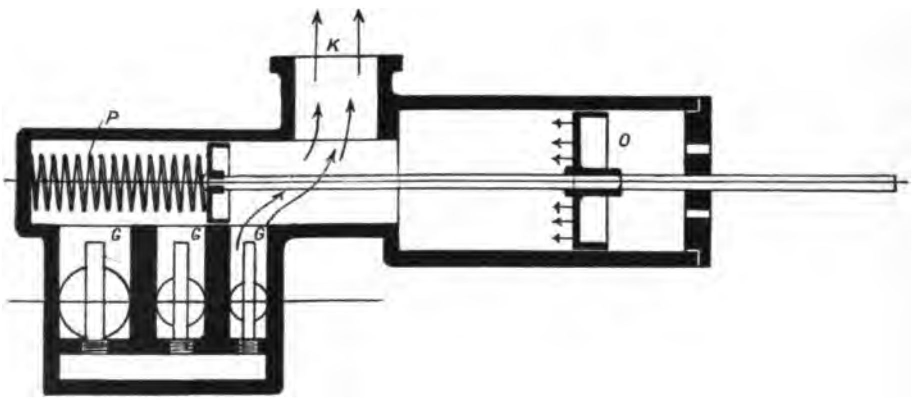


Fig. 92.

G, G, G, Air nozzles of different sizes, each with a suitable petrol jet.

As the engine sucks harder at K, the piston, O, is sucked along against the spring, P, and opens the larger nozzles successively.

that the actual working of a carburettor seems to depend mainly on the careful adjustment of the various parts. The right proportions may be determined by abstract calculation, but the method of trial and error seems to be the easiest way of obtaining good practical results. The simplest way of doing this is probably as follows:—

Adjustment of Carburettors.—Find by experiment what is the largest air nozzle with which the throttled engine will run smoothly and satisfactorily at a slow speed. Arrange the automatic adjustment so that it can be worked by hand, and provide means for measuring the vacuum produced. Then, by running the engine at increasing speeds, experimentally determine the amount of adjustment necessary to get the best results with each different rate of flow through the carburettor, and also the vacuum which will effect the adjustment. Then arrange the springs, &c., so as to get the same adjustment at the same amount of vacuum. It is obvious that the larger is the nozzle that works satisfactorily at the lowest rate of

flow, the easier will be the adjustments for increased rates, and the less the throttling.

In practice, the air nozzle is made about a sixth to a fourth of the diameter of the cylinder. If the nozzle is about a fifth of the piston area, there will be an extra air supply, at least corresponding to the area through the nozzle. The rate of flow will then be about 12,500 feet a minute, when the engine is running at a piston speed of 1,000 feet a minute with the extra air valve full open, and about 800 feet a minute when the engine is running slow and throttled, with the air valve quite shut.

Vaporisation.—The effective vaporisation of the petrol depends very largely on there being a rapid current of air past the jet. To obtain this with the least throttling of the charge, the passage ought, theoretically, to

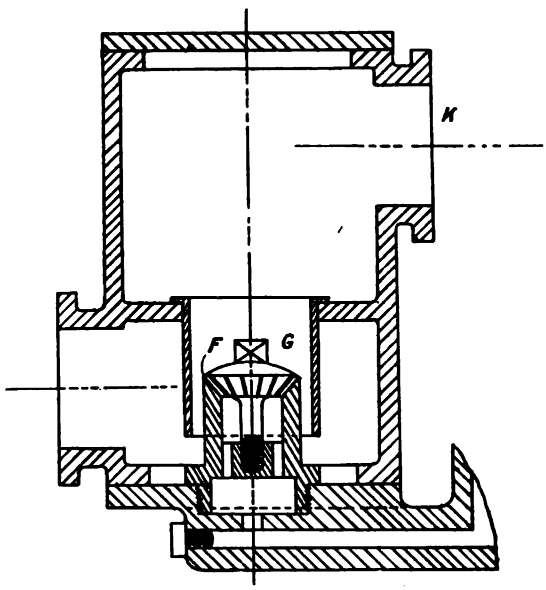


Fig. 93.

F, Petrol jet. G, Air nozzle. K, Inlet to engine.

be gradually contracted and expanded as in fig. 83, and as is done in some carburettors. The finer the spray, the quicker is the evaporation; hence a multiple jet, like that shown in fig. 93, is sometimes adopted; but this does not seem to be sound practice, as small jets are more easily choked up than a single large jet; and are more difficult to make absolutely alike, and, owing to capillary attraction, will not be effective over so extensive a range.

The petrol is, generally speaking, better sprayed by a transverse than by a concentric current of air; in the latter case, the hole is sometimes placed at the side of the jet instead of at the end, but this makes it less easy to see whether it is clear. The simplest way of making the petrol spray well from a single jet concentric with the air nozzle seems to be to put a flat-ended rod just over it, as in fig. 89. Two jets impinging on each other have also been used.

Although petrol will evaporate completely in air at the ordinary temperature, the evaporation may cool the carburettor below the evaporating temperature. To guard against this it may require to be slightly warmed. Generally speaking, the carburettor is placed in the bonnet of the car, where it is surrounded with air which has come through the radiator and is, therefore, somewhat warm. This is often sufficient, but if more heating is desired the air may be drawn from near the exhaust pipe, which will, therefore, be warm, or that part of the carburettor may be water-jacketed (figs. 83 and 88). The latter plan is to be preferred, as the petrol should be evaporated from a hot surface rather than by hot air, and the evaporation of the petrol helps to cool the water.

Petrol Level.—It is commonly supposed that it is absolutely necessary to have the level of the petrol and the jet so arranged that the petrol stands just at the top of the jet. In practice this seems very doubtful. In most cars it will be found that if the petrol is shut off while the engine is running and the engine is left running, the fall of the level of the petrol in the float

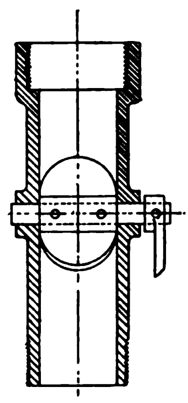


Fig. 94.

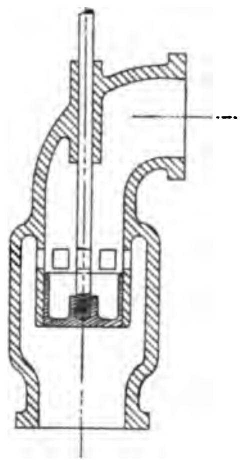


Fig. 95.

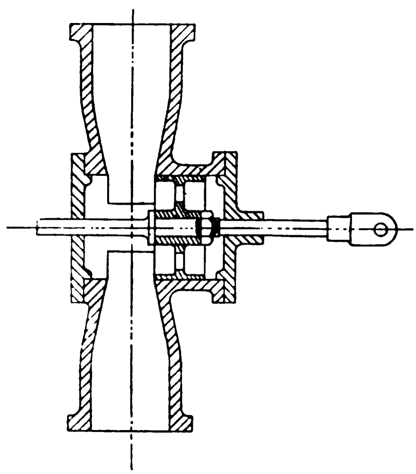


Fig. 96.

The valve is not balanced in this case as in fig. 95; but if carried on the spindle it works very easily.

chamber makes no difference to the running of the engine till it is quite exhausted and the engine sucks air through the jet. If the petrol was so arranged that it stood some way down the jet, say ½ inch, it would be much less liable to overflow in case of the float not closing immediately the level of the petrol rose in the float chamber and would in fact give more margin. It seems that it is only a matter of adjusting the air supplies, and this is often effected automatically.

Practical Points.—Carburettors should be easily accessible; there should be no difficulty in getting at the petrol jet; the parts should be few, sufficiently strong, and easily fitted together. The whole should be so arranged that there are the least possible settings to machine and as few delicate parts to be fitted together as possible. The carburettor should be conveniently

carried on a machined facing on the engine or be bolted on to the inlet facing. The float chamber should be easily accessible and the float easy to take out without removing any pipes or other parts; and the lid of the float chamber should be fastened with a bayonet joint or with only one or two screws of reasonable size. A lot of very small screws are a nuisance and expensive to make.

It would be a convenience if the petrol jet had a thread corresponding with that of the tyre pump, as then it could be thoroughly cleared by blowing air through it.

In throttle-governed engines the throttle is most conveniently made part of the carburettor. If carefully arranged the machining to it will be done at the same setting as some of the other machining. The general types of throttles are the butterfly valve (fig. 94) and the piston valve (figs. 95 and 96). The latter seems the best in many ways, as the former is not quite balanced when there is a rapid flow through it, and, therefore, will want a slightly more powerful governor. Also, unless the pipe is jointed across the line of the spindle it is not quite so easy to fix the valve to the spindle in a satisfactory way. The piston type, if properly arranged for being machined at the same setting as other parts, should be no more expensive.

CHAPTER VI.

CYLINDERS, PISTONS, VALVES, &c.

Cylinders.—The design of the cylinders requires special care as these constitute the important parts of the engine, as (1) the power is produced in them, and (2) the arrangement of the cylinders and their valves entirely governs the whole arrangement of the engine. Theoretically, as regards economy and power, the combustion chamber of a cylinder should be a sphere, as it is the most compact form and has the least cooling surface in proportion to its volume, but is not a convenient shape to make, an approach to it can be made by putting the valves in the head of the cylinder and at an angle. This should give the greatest possible power, because the smaller the surface for a given volume the less heat is lost to the water jacket, and because the more compact the combustion chamber, the more nearly will the combustion approach instantaneousness.

In practice the shape of the combustion chamber is, as a rule, entirely settled by the arrangement of the mechanical parts of the engine, as it is far more important to make an engine that will run satisfactorily, than one that will either develop the maximum of power for a given cylinder or that saves a few shillings worth of petrol. Still, in comparing the different arrangements of cylinder it is as well to keep in mind the essentials of an ideally perfect engine, and for special cases, where lightness or economy is of exceptional importance, it may be well to modify such ideas in this direction.

It might be thought also that less compression might be required to develop the same power from an engine having a compact combustion chamber than from one which has a more straggling shape of chamber. The difficulty of developing the power of an engine when the compression is very low is mainly due to the combustion being too slow (see *Compression*), while the advantages of low compression are smoothness of running, ease of starting, small wear on the bearings, and less strain on the parts.

Valves.—In practice cylinders have the following different arrangements of valves, the order adopted being that of their approach to the spherical shape, which is, theoretically, the right one. These are illustrated in figs. 97 to 103,* all of which are drawn to the same scale for the same sized cylinder having the same proportion of compression space; they show what surface is exposed to the hot gases :—

Fig. 97 is not much used, as it is difficult to make a satisfactory valve

* Figs. 97 to 103 show combustion space of cylinder having compression ratio ·33 of volume swept out by piston, and stroke equal to diameter.

Relative surface of combustion chamber, taking fig. 97 as 100.

Fig. 97,	100	Fig. 101,	148
,, 98,	112	,, 102,	190
,, 99,	140	,, 103,	180
,, 100,	148		

gear, and it is also very inconvenient to machine. The valves being oriented at various angles, the seatings must not only be machined at different settings but also at various angles. The real difficulty is the valve gear. Either the cam shaft must be along the top of the cylinders or the driving must be effected by a complicated arrangement of rocking levers. While it has more disadvantages than the next arrangement, it will allow of the ignition plug being neatly placed between the valves on the top of the cylinder.

Fig. 98 is but little more used than the last. The shape of the combustion chamber is almost as good as in the preceding example, while it is

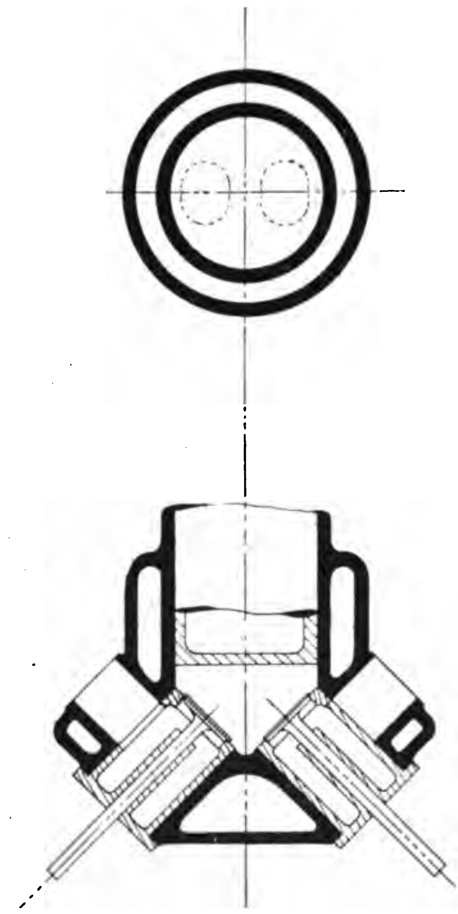


Fig. 97.

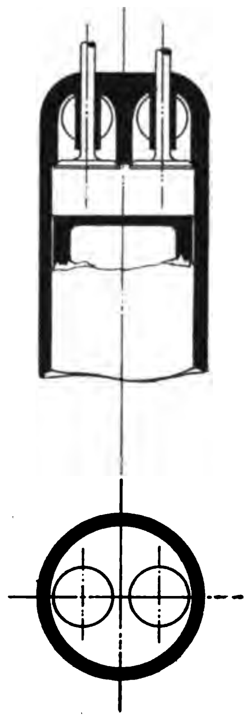


Fig. 98.

more convenient to machine, as both the valves are parallel with the cylinder. For the same reason the valve gear is probably more easily arranged, though here also either the shaft must run along the top of the

cylinders or rocking levers are required. If used for small engines there is a difficulty in finding a convenient place for the ignition plug and sometimes in arranging the trip gear for a low-tension ignition. It is almost absolutely necessary to put the ignition in the side of the cylinder above the reach of the piston, and then the oil is more likely to get on it than when it is in a pocket, by the inlet valve, or in the centre of the cylinder.

In the arrangement shown in figs. 97 and 98 the seatings of the valves must be loose, which slightly increases the expense as compared with that of casting seatings and valves with the cylinder, but this is compensated for by the seatings being renewable. The matter is not of much importance, as the seatings of the valves generally last as long as they are wanted, but in commercial work, where cars are run much more continuously than pleasure cars, it may be a consideration.

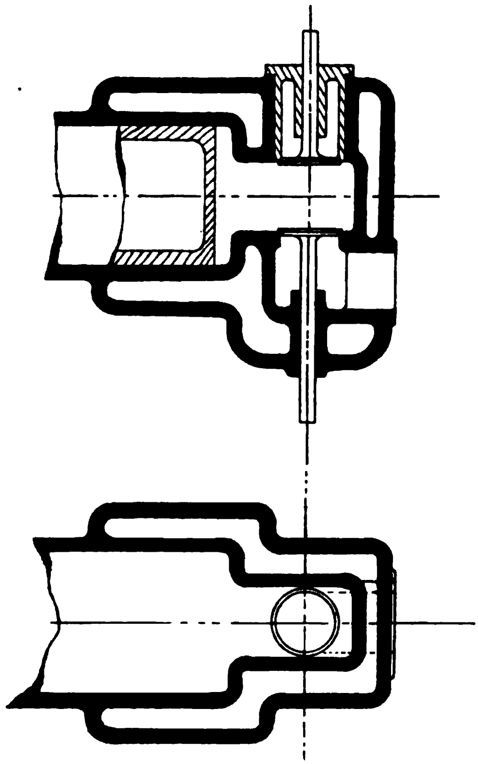


Fig. 99.

Fig. 99 is more often used for horizontal cylinders than for vertical, but has been used with success in the latter. The combustion chamber is good in shape and it is cheap to machine, as the inlet and exhaust, being on the same centre, can be machined together. It has one loose valve seating only, instead of two as in figs. 97 and 98, but there is the same difficulty with the valve gear as in those just referred to. Other arrangements are possible, but cost more than the plans next described in which the valves are vertical.

Other arrangements with the valves at right angles to the cylinder are used in horizontal engines, but their advantages and disadvantages will be understood from the discussion on vertical valves in vertical engines.

Fig. 100 shows an arrangement which was very common when inlet valves were usually automatic. Its great advantage is its cheapness, as both the valves are on one centre, and, like fig. 99, it has one joint only for both valves. This is a distinct advantage in small engines, as joints are always apt to leak. In removing and replacing both valves one joint only need be broken and remade. All the valves and low-tension ignition can be arranged to work from one cam shaft. The pipe arrangement also is good, as the carburettor can be placed at the opposite side of the engine to the cam shaft and the inlet pipes be conveniently carried over the tops of the cylinders. The ignition can be conveniently placed between

the valves, well away from the chance of getting oil on it, and easy of access. The combustion chamber, though theoretically not quite so good as in the preceding instances, gives good results in practice. One advantage of this and fig. 99 is that the passage from the valve chamber to the cylinder can be made of such a shape that if a valve head breaks off the stem it is impossible for it to get into the cylinder. Should this happen its extraction is difficult.

The disadvantage of this arrangement is that if a mechanical inlet is wanted it must be worked by a rocking lever.

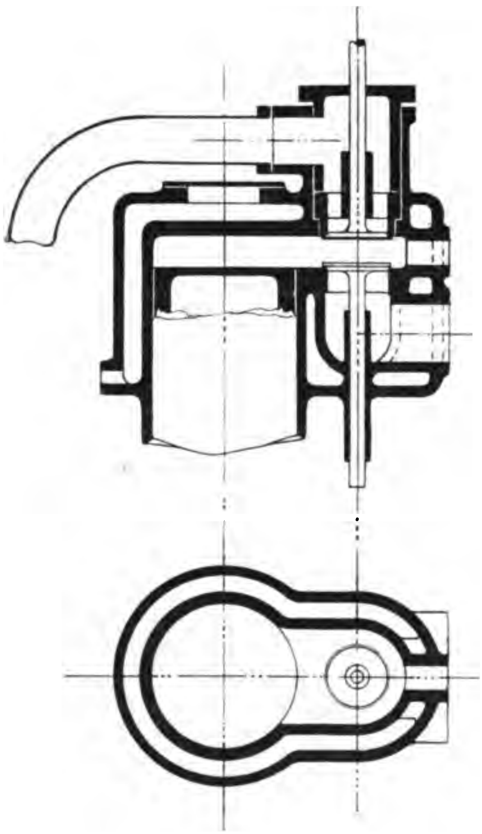


Fig. 100.

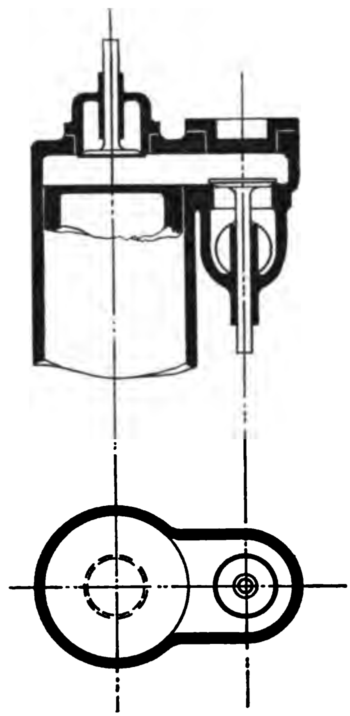


Fig. 101.

Fig. 101 is an arrangement with practically the same shaped combustion chamber as the last. Its disadvantages are that there is an extra joint, an extra seating to machine (as the valves are not on the same centre), and that the inlet valve has to be worked by a rocking lever, if it is mechanical. It seems to be inferior to the others, and its temporary popularity was fictitious.

The arrangement shown in fig. 102 is very largely used, though not so much as the next. It has both valve seats cast in the cylinder and is fairly

cheap to machine, as both are alike and both vertical. Both valves can be worked without rocking levers, so that the expense of these is saved. All the gear can be arranged on one cam shaft, though careful designing is required to make it satisfactory with low-tension ignition. One disadvantage is that if large valves are used it makes rather a long engine, particularly when the cylinders are cast separate, as the valve chamber is then considerably wider than the cylinder. Its main drawback is the lack of facility for arranging the pipes nicely and for putting the low-tension ignition on the same cam shaft as the valves. The inlet and exhaust pipes are both on one side, and naturally come out of the cylinder at about the same level. If they are deflected directly downwards they are inconvenient for getting at the valve springs; if, on the other hand, the carburettor is placed at the back,

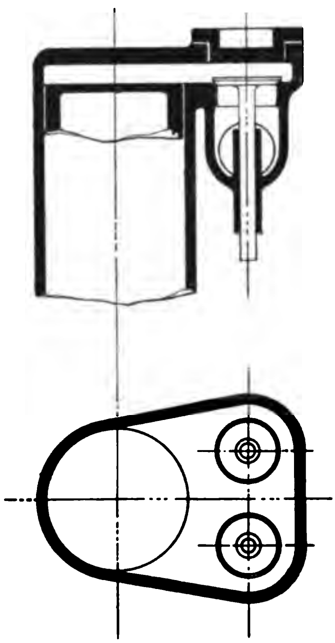


Fig. 102.

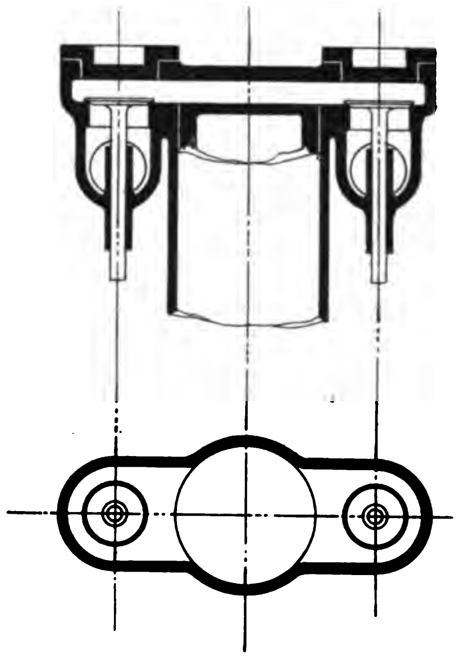


Fig. 103.

the inlet pipes are long. Generally the pipes are superposed castings suitable for making a neat arrangement.

Occasionally the low-tension ignition is placed on the other side of the cylinders, but this entails an extra cam shaft, and, as the plugs are just over the piston at the side of the cylinder, they are liable to be oiled.

Although the arrangement shown in fig. 103 has the worst shaped combustion chamber it is the one in most general use, as it is very convenient and is easy to design. The pipe arrangement is good, as the exhaust is placed on one side and the carburettor on the other. The centres of the cylinders are readily placed near together, as the valve chambers are smaller than the cylinders.

The chief objections to it are the bad shape of the combustion chamber,

the expense of having two cam shafts, and, sometimes, that the engine has to be reached from both sides.

In the arrangement shown in figs. 97 to 100 there are no caps to the valves, as the latter, with their seats, close all the openings. In the others, on the other hand, one or two of the valves have covers which it is not convenient to water-jacket. This is not a great matter when the valves are small, as the cover will be fairly well cooled by contact with the water-jacketed cylinder; but in larger engines its size will be a disadvantage. It is best not to have any unjacketed caps, the plan adopted in the first four arrangements.

Caps with holding studs are a source of expense, and, although in figs. 97 to 100 there are loose seatings to the valves, these will not be more expensive than the caps to the valve holes in figs. 101, 102, and 103. In some cases, however, the cap of the valve hole is used as the ignition plug, and this may save there being a special plug for it.

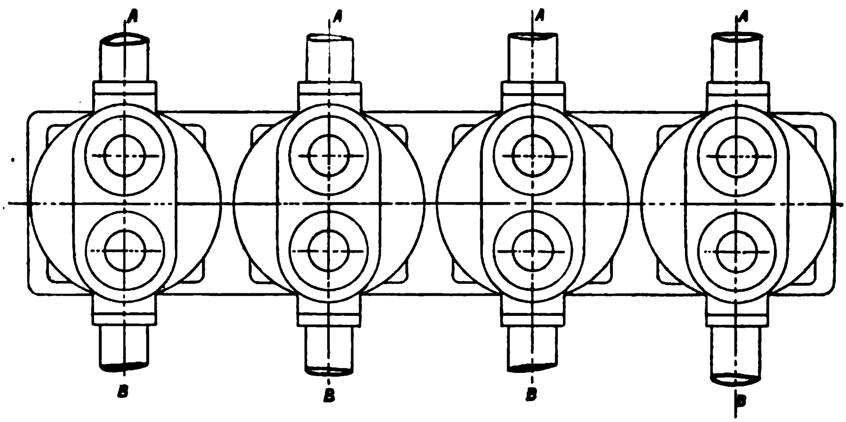


Fig. 104.

A, Inlet pipes. B, Exhaust pipes.

This makes a much better pipe arrangement than placing all the valves on the line of the cylinders, but the valve gear is very difficult to arrange nicely.

In most of these arrangements the valves are interchangeable, as the seatings are alike. In figs. 100 and 101, however, the seatings are not alike, though the valves can be made so if desired.

It is very important that the arrangement of the pipes be such as will allow of ready accessibility to the engine. If the valves are at the top of the cylinder, whether inclined or vertical, it is generally easy to manage this, particularly in the larger sizes. In the former the pipes will issue from opposite sides of the cylinder, as in fig. 97. In the latter the valves may be placed either along the centre line of the engine or crosswise (fig. 104). In the former case the pipes will all come out on one side of the engine, and the valve gear can then be conveniently placed at the other, as fig. 23.

Very good engines have been made with all these arrangements, and it is simply a question of which gives the best result for the least money. Personally, I prefer the arrangement shown in fig. 100, and to avoid having two cam shafts. The expense of an extra cam shaft is considerable, as there

is not only the shaft, but also the bearings it runs in and the gear wheels that drive it. It is true that in fig. 100 the inlet valve has to be driven by a rocking lever, while in fig. 102 this is done away with; but the latter has an extra cap over the inlet valve which will cost nearly as much and a worse shaped combustion chamber, while the pipe arrangement is inconvenient, and may require the whole engine to be longer, thereby making the crank shaft and crank case heavier and more expensive. There is, however, so much to be said for and against each arrangement, that the choice must be very much a matter of individual opinion and dependent on the purpose for which the engine is used.

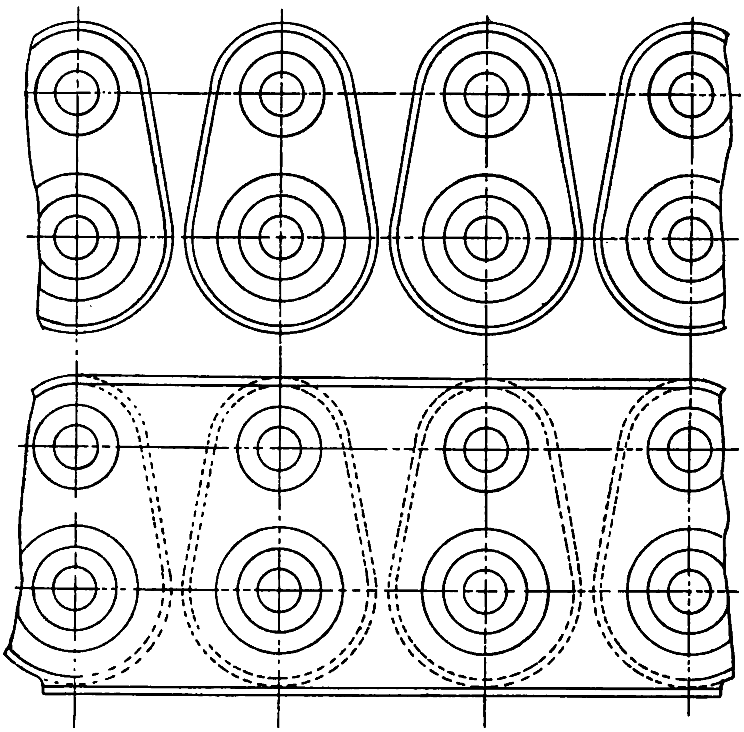
Casting of Cylinders.—The next point to be considered is whether the cylinders should be cast separately, in pairs, or in one piece. The plan most favoured at first was to cast the cylinders in pairs, because two-cylindered engines were in general use, later four-cylindered engines were made, which was most easily done by combining two pairs. On the other hand, most of the early light car makers used separate cylinders because the patterns they followed were single-cylindered engines, and if more power was required they simply used a larger number of single cylinders.

As regards the use of cylinders cast in pairs, this partly depends on whether it is intended to have a bearing between each crank, or to have the adjacent cranks of each pair without one. This matter will be gone into more fully under the heading of cranks, but it appears to me that if no bearing is used between the cranks, the cylinders should certainly be cast in pairs. By doing this it is possible to bring their centres a great deal closer than if they are cast separately. At the same time, a good long bearing should be put between the middle cranks, if the engine is four-cylindered; and this throws the centres rather far apart to cast them all in one piece. On the other hand, if there is to be a bearing between each crank (which I much prefer myself), the centres of the cylinders will be at even distances, and it will thus be more convenient to cast them all together or all separately.

The advantage of casting each cylinder separately is that a single cylinder can be renewed when damaged, or rejected, if there is a flaw in the casting. On the other hand, any defect in one cylinder involves the whole casting if they are cast together. The objection has no great weight as cylinders should rarely be damaged, and with good foundry work it is quite exceptional to have wasters, and if it be disregarded it is certainly the most advantageous plan to cast all the cylinders in one piece.

Fig. 105 represents four cylinders as cast separately, and fig. 106 the cylinders as cast in one piece, and the amount of cylinder wall saved is indicated by dotted lines. It will be seen that this is considerable, and, therefore, the cylinders all cast in one will be a good deal the lighter. Besides this, only one inlet and one outlet for the water circulation will be needed, and this will save a good deal. Another point is that one long door (which is lighter than the cast jacket) can serve as the back of the water-jacket and allow of the core being taken out. With separate cylinders each cylinder will require a door, and some have four doors, for the withdrawal of the core; this makes 16 doors for a four-cylinder engine, which adds greatly to the expense. As regards cost, the cylinders cast in one will probably be a good deal the cheaper, provided that they are made in equal quantities. In suitable machines the casting can be bolted down and all the cylinders bored at one setting, the casting being moved along

on the boring machine; or even separate spindles may be used so that all are bored at once. There is also the saving from having one door for removing the core and only one set of water connections. The pipe work connecting the water circulation to all the jackets is a very expensive item in engines with separate cylinders. In some designs there is no reason why the inlet and exhaust passages to the separate valve chests should not also be cast in and pipe connections to them avoided. Fewer holding-down bolts will also be required. On the other hand, if all the cylinders are cast



Figs. 105 and 106.

in one it is obvious that special patterns will be required for each number of cylinders. If they are all cast separately, then one stock cylinder can be made, which can be used for engines with one cylinder or any number. If the engines have varying numbers of cylinders one pattern can be multiplied if the same pattern does for all.

The matter is, therefore, rather a commercial one. If a firm is going to specialise on engines with a given number of cylinders, say four, then it will probably be best to cast them all in one. If, on the other hand, they make sets of engines with varying numbers of cylinders it may be best to cast the cylinders separately.

When engines are made on a very large scale, however, it may be worth while having special patterns for each number of cylinders, and making the working parts only of the engine of the same pattern.

Casting the cylinders in pairs is a compromise, and like most compromises it does not have the complete advantages of either plan, and has a good deal of the disadvantages of both. It is, however, still the most used and probably has a good deal in its favour. It allows the same stock pattern to be used for engines of two, four, or six cylinders, while special patterns have to be made for the one and three cylinders only. In a plan I have used (shown in fig. 22) an endeavour has been made to obtain most of the advantages of both plans. The cylinders are cast with a large opening, as in fig. 112, and bolted up together. One water jacket suffices for all the cylinders, while the large opening facilitates the removal of the core, whereas if the casting has small holes this operation entails great labour. Further, a small amount of sand is liable to be left in the core, and, if this works into the pump, may cause a great deal of unnecessary wear there. In some cases the jackets are cast with several holes in them, which are closed with plugs after the core has been extracted. This is rather apt to be expensive, and not very satisfactory. Many cylinders have an opening at each side similar to that shown in fig. 112, but

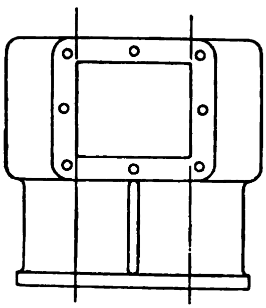


Fig. 107.

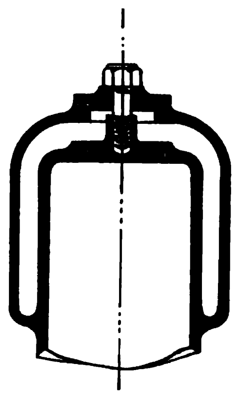


Fig. 108.

closed with a door so that each cylinder has its own jacket. If the valves are on one side and the cylinders are cast together a large door at the back of the cylinders (fig. 107) is probably the best. If the valves are at the top of the cylinders, such a door can be put at either side, and, as this may be of thin aluminium, the weight saved as compared with a cast jacket all round is considerable.

In single cylinders an alternative to fig. 112 is to have a large door at the top, as shown in fig. 100; or, as is often done, to cast a hole in the top of the cylinder through which a boring bar can be passed and the core easily removed. It is not really necessary, as the cylinders can quite well be bored without, but is perhaps of sufficient advantage to be worth doing. It is generally made as in fig. 108. There is a slight disadvantage in that the leakage from a joint being into the water jacket, it is, therefore, not quite so apparent as a leakage into the open air. Another is, that if water leaks into the cylinder, the ignition plug may be short circuited. These are of little practical importance, but are avoided by making the cylinder in the way shown in fig. 109. In this case, however, the hole at the top of the cylinder

cannot be used for the extraction of the water-jacket core, which must be effected in some other way.

In some of the older engines the cylinder head and barrel were cast separately, and a joint made across the jacket, so as to serve as an inside and an outside water-joint, but owing to the difficulty of making the joints tight it has been abandoned.

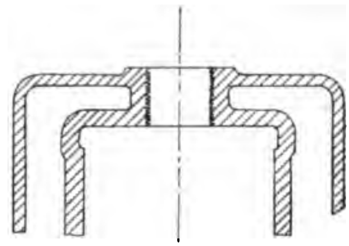


Fig. 109.

A plan of making the water jacket, which has some advantages, is shown in fig. 110. The jacket itself is part of the crank case, and the cylinder is a liner slipped inside it. It makes the cylinder a very simple casting, and the core of the jacket round the head can be got out very easily, while the cylinder can be turned all over the outside of the barrel. It can, therefore, be made a little thinner than it otherwise could be, as there need be no allowance for unsound castings or uneven thickness, &c. On the other hand, it entails two water joints, and it is more expensive than making the cylinders all of cast iron, as the crank case has to be bored to each cylinder centre, and the cylinder has

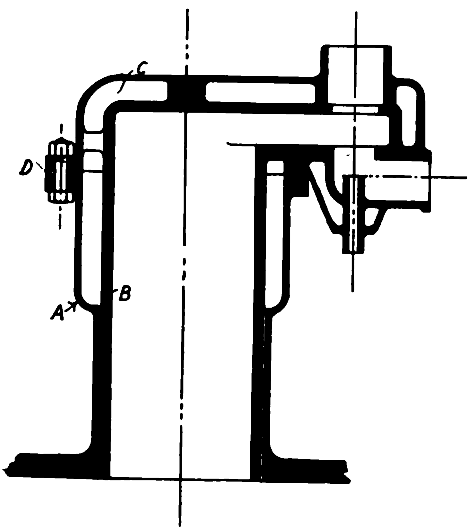


Fig. 110.

The casing of the water jacket, A, is made of aluminium, and forms part of the crank case. The cylinder, B, and head of water jacket, C, are cast in one, and bolted to A at D.

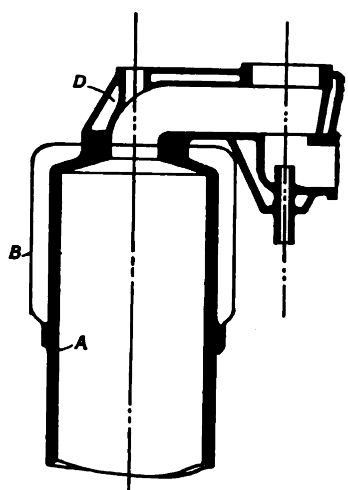


Fig. 111.

A, Steel cylinder.
B, Jacket of copper sheet.
D, Casting, forming cylinder head and valve chamber.

also to be turned to fit it. In this arrangement the whole of the jacket of the lower part of the cylinders can be made in one piece, so that only one inlet pipe is required; but the tops are all separate, so that there must be an outlet pipe from each. It has never been much used, and has been generally abandoned by those who adopted it.

Another plan that has had a certain vogue where weight is of great importance, as in racing engines, is to have the cylinders steel tubes, and make the jackets of brass or copper sheets. One arrangement of this is shown in fig. 111. There are several different ways of arranging the jacket, but the above represents fairly the principle on which most of them are made. This plan has not come into use as much as was anticipated

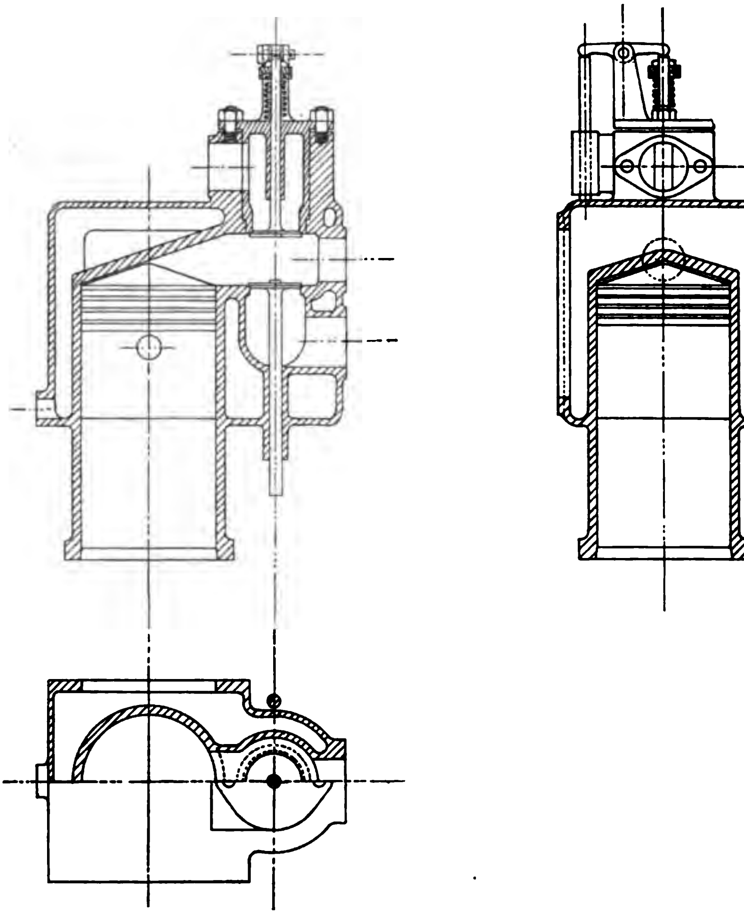


Fig. 112.

when it was introduced some years ago. That a lighter cylinder can be made this way than by casting the jacket all round it is obvious, but, on the other hand, it practically entails having the cylinders separate, though a brass or copper jacket is sometimes put round a pair of cylinders. This generally involves a great many water connections, as the jackets are, in most instances, separate from the heads, and, therefore, there are practically two jackets to each cylinder. This means eight inlets and eight outlets for a four-cylinder engine; the weight of these almost equals the weight saved

by the thin jackets and cylinders. Again, the engine has to be rather longer than when the cylinders are cast, particularly if they are cast together, and this lengthens the crank case and crank shaft, and also the frame of the car. It is probably for these reasons that it has not been very successful, even in racing cars, the large majority of races lately having been won by cars with cast cylinders.

For anything except racing, cast iron is the best, and by far the cheapest, material for cylinders.

Water Jackets.—It is usual to carry the jacket down the cylinder barrel as far as the bottom of the travel of the piston, but it is doubtful if this is necessary (especially where weight is of great importance), as the heat just at the bottom of the stroke cannot be great, and is easily carried off if the water jacket is near. On the other hand, the head should be very thoroughly jacketed, as the heat here is very great, and the head is exposed to it during the whole of both the working and exhaust stroke. It is well to carry the jacket well above the top of the cylinder, and to be careful that there are no corners or spaces for the lodgment of steam, and that it is so arranged that steam cannot lodge in it when the engine is inclined on hill slopes. The jacket should also be carried well rouud the valves.

Fig. 112 shows a cylinder which will illustrate some of these points. The valve is here placed over the exhaust, but the details will serve for any other type.

The distance of the exhaust valve from the centre of the cylinder will, in practice, be settled by the fact that there should be a water jacket between it and the cylinder. It may, however, be desirable to carry it out further than this, in order to suit the design of the crank case, but the less this can be done the better, as it makes a badly shaped combustion chamber. The valve should have a clear way all round the head (as shown), in order that its full area may be utilised. If the wall of the valve chamber comes too close to it, this cannot be done. It is probably better to thicken the exhaust valve seating (as shown), in order that it shall not be cracked or distorted.

The cylinder (as shown) is arranged for low-tension ignition. Should high tension only be used, the plug would best be placed where the low-tension one is shown. Should both high and low tension be used, the best place for the high-tension plug would probably be at the top of the cylinder, and in the middle. In this case the inlet pipe must be shifted a little out of centre, or be brought in from the other side.

If the valve is inverted and carried on a loose seating, it may be of the form shown, with the spring outside; or it may be arranged as in fig. 100, with a loose seating held down by a cap with the spring inside. The latter is the usual arrangement for automatic valves, as they can then be made both much shorter and lighter. The former is used if both the exhaust and inlet are inverted, as the spring must then be outside.

The seating and general arrangement of an automatic valve are shown in fig. 113. The stem forming the valve guide is carried on bars, which are formed by casting the seat solid and drilling holes. Cast bars may be slightly cheaper, but the plan shown has the advantage that it can be seen that the casting is sound. The seating should not be too shallow, if it is a loose one, as shallow-seated valves cannot be screwed up tight, their seat being liable to be twisted under the unevenness of the packing, and the valve itself to become leaky.

The guide of the exhaust-valve spindle is shown cast in the cylinder. This construction is coming largely into use, but many are still made as in fig. 114, the guide being a loose piece screwed into the cylinder. The idea of this was, no doubt, that it could be renewed when worn, but it is considerably more expensive, as the guide has to be made a separate piece, and, also, the hole has to be tapped and faced from the under side, entailing an additional setting. The cast guide can also be water-jacketed, which keeps it cool and helps to keep the exhaust valve cool. It is not found that there is any material wear on these cast guides, but, if they should wear, they can easily be bored out and bushed, and, if desired, it would be quite possible to put a bush in them to start with.

If both the valves are worked direct, the inlet valve will be similar to the exhaust, and, therefore, there is no need of further description. If both

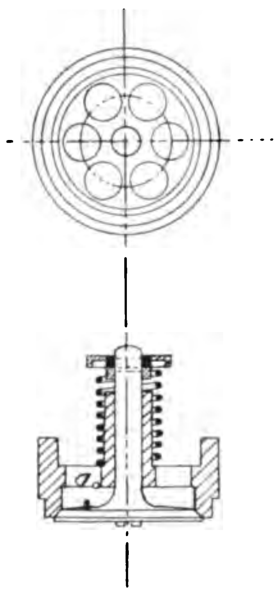


Fig. 113.

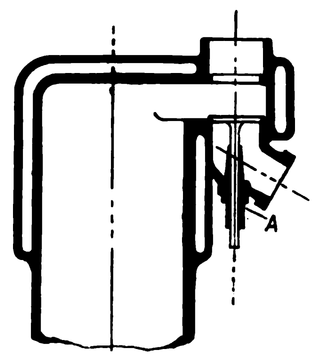


Fig. 114.

the valves are inverted, the exhaust will be similar to the inverted inlet. It is generally convenient, in the latter case, to carry the water jacket right up to the top of the valve cap and cast the pipes through it, but this is not necessary.

A point that should be carefully considered is the advisability of screwing caps into cast iron, either to secure the valves or to close the holes through which they are inserted. In some designs of engines this can hardly be avoided, as in the case of small engines with all the valves in line along the top of the cylinders. Inlet valves are sometimes held in place by such caps as are shown in fig. 115, and, if both valves are direct, caps are screwed in to cover them.

In most kinds of engineering this is not considered good practice, as the threads in cast iron are liable to crumble when the caps have been screwed

in and out several times; the usual plan is to hold all parts on with studs or bolts, which are not so liable to damage and can be renewed when worn.

If the valves are inverted, the lever working them is sometimes passed through the valve cap, as shown in fig. 116. This seems to have no advantage over the plain lever; it is more expensive and has more parts to work loose.

In designing cylinders, the parts should be so arranged as to require as few settings as possible. For instance, the exhaust pipe shown in fig. 114 is at an angle different from anything else, and therefore has to have a special setting to machine it. By setting it vertically, as in fig. 112, it can be machined at the same setting as the ignition plug and other parts. A little forethought in these matters will materially lessen the cost.

Horizontal Cylinders.—Horizontal cylinders are not much used in European motors now, but will possibly be very general in the future. The

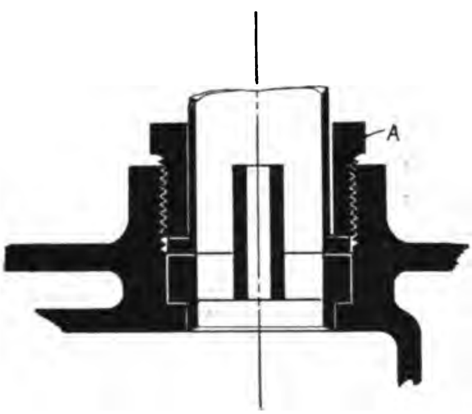


Fig. 115.
A, Cap screwed into cylinder to hold down inlet-valve seating.

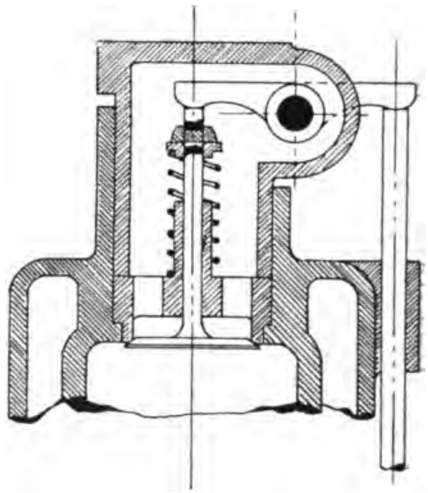


Fig. 116.

vertical arrangements above described can be used by simply laying the cylinders on their sides. In this case the valves will work horizontally, except in the arrangement shown in fig. 100. Horizontal valves are much favoured in gas engine practice, and should be equally good for motor cars. If, as is usual, they are vertical, they can be placed either over each other or side by side, either at the end or side of the cylinder.

Diagonal engines, with the cylinders working on one crank at right angles, are also useful for some purposes, and may be extensively used for marine work, as they have the advantage of being very low for their power. They also make very cheap engines for light cars and tri-cars, as the crank can be built up (as in the case of the single-cylinder engine), while the connecting-rods and main bearings can be bushed on hardened pins. This type of engine is usually made by simply inclining a cylinder as made for a vertical engine. For small engines this is a very cheap and, probably, a satisfactory arrangement.

If four or more cylinders are used, an accessible arrangement cannot be thus made, and one that might be a considerable improvement on it is shown diagrammatically in fig. 117.

Another plan for a low engine is shown in fig. 118.

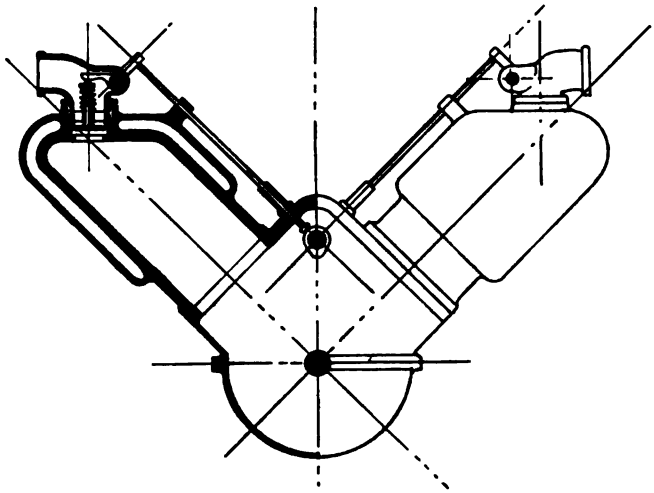


Fig. 117.

With regard to the proportions of cylinders as to strength in small engines, this is usually settled by the thickness that can be satisfactorily cast. It is not easy to obtain a uniform casting with cylinders less than

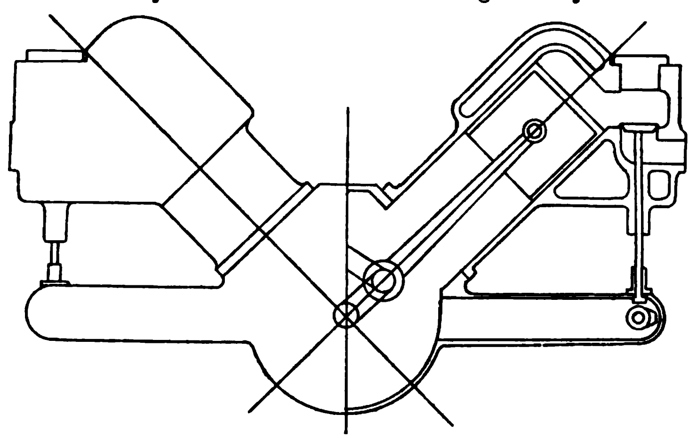


Fig. 118.

$\frac{1}{4}$ inch thick. In the larger engines the metal may be, roughly speaking, one-fifth of the diameter of the cylinder. This will allow a stress of about 2,000 lbs. per square inch on the metal with moderate compression. The top of the cylinder must be thickened to allow for the strain there, though,

in the engines of the size usually put into a car, metal from $\frac{1}{4}$ to $\frac{3}{8}$ inch stands perfectly.

Sometimes the valve seats in cylinders are sunk in a well (as in fig. 119) instead of being made level with the port (as in fig. 112). Two reasons are given for this. The first is that, if the valve seat is level with the bottom of the port, the rush of gas through the latter will put a side pressure on the valve and tend to make it wear the guide on one side; but there is no support for this in practice, as valves have shown no signs of this after working for years. The other is that, when broken off, the head is less likely to go into the cylinder. This is true, but is easily prevented in other ways.

The cylinders will require a base for the attachment to the crank case. Generally there are four holding-down bolts when the cylinders are cast separately, but only two between each pair if they are cast together. This means that in a four-cylinder engine with four separate cylinders there are 16 bolts; whereas, if the cylinders are cast in one piece, only 10 are needed. In the latter case, also, all the bolts will take a certain amount of the strain of the explosion, whereas, in the former case, it is concentrated on the bolts holding down each cylinder. Still it is safer to assume that it may come on the four bolts adjacent to each cylinder, even if the four make one casting. These bolts are generally made about $\frac{1}{6}$ to $\frac{1}{8}$ of the diameter of the cylinder, which allows of a strain of about 5,000 to 8,000 lbs. per square inch, allowing for a maximum pressure in the cylinder of 300 lbs.

When convenient the bolts which hold down the cylinders should be taken right down to the bearings to hold the caps on, so as to avoid any strain on the crank case. This is most easily managed when the cylinders form one casting.

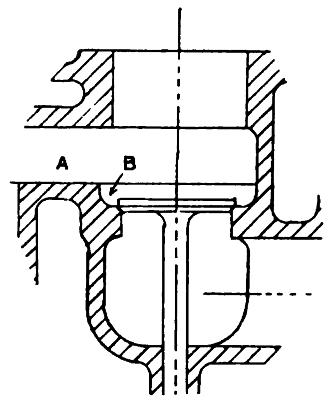


Fig. 119.

A, Port to cylinder.
B, Sunk valve seat.

Pistons.—These should be as light as possible, consistent with reliability and durability. The usual material is cast iron. These can be made light enough for use in all but the fastest engines. Steel pistons can be made lighter, but as the surface on these, both inside and outside, has to be machined the cost is very much greater.

The length of pistons has varied from twice the diameter of the cylinders in the early low-speed engine to a length equal to the diameter of the cylinder in the high-speed engine. The long piston had the advantage that there was more surface to take the wear, and was less liable to twisting in the cylinder; but it had the disadvantage of its greater weight, and required the engine to be higher and heavier. Pistons with a length, not less than the diameter of the cylinder are found to wear well; and the lighter they are the less is the wear on the brasses. This is now the usual proportion adopted.

Fig. 120 shows a very satisfactory design for a piston to work in a 4-inch cylinder. With the rings and pin it weighs just 3 lbs. It could be made somewhat lighter, as is sometimes done for racing cars, by the lower part

being bored so to leave only just enough metal to guide the piston; but the gain is so small that it is not worth doing for ordinary work.

The piston is kept tight by spring rings, of which there are three in the case shown above. These are always of cast iron.

In gas engines which run continuously for long periods, six or more rings

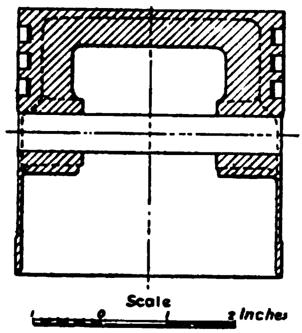


Fig. 120.

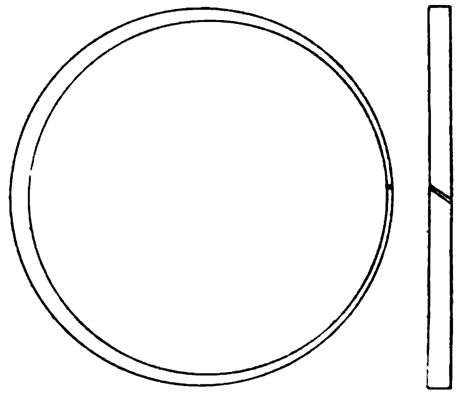


Fig. 121.

are usual; but this would make the pistons too heavy for petrol motors, therefore fewer are used at the risk of repairs being needed more frequently. It might be better to have four rings and a heavier piston, but this would depend on the work required from the engine as regards speed and duration.

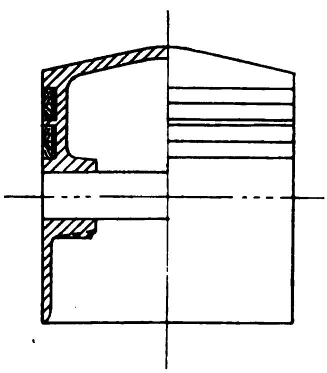


Fig. 122.

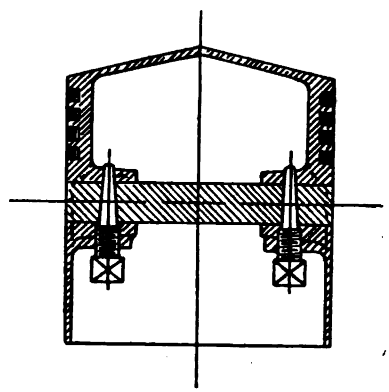


Fig. 123.

Piston Rings.—Rings are made, as in fig. 121, thicker at the root than the joint and with a slot at an angle. It is very important that this size should be such that the slot closes up when they are in place, as otherwise there will be a leak there. The diameter of the ring exceeds that of the cylinder by about one-thirtieth before it is sprung in; in order to make it a good fit it is often compressed and ground after being turned. This answers well, but is not really necessary, as rings turned and then put in are soon

worn down so as to fit satisfactorily. In some cases the piston has four rings in two grooves, as in fig. 122. This does not seem any better than two rings, as the gas goes round the back of the rings.

In any case it is better to have several narrow rings rather than a few wide ones as the leakage is mostly at the joint of the ring, and, therefore, the more rings there are the more the gas is impeded.

The top of the piston may be either strengthened with a web across it, as in fig. 120, or it may be domed or coned up, as in fig. 123. The latter is slightly the lighter as there is no web, and it has the advantage that any oil put into the cylinder will immediately run down the sides. The figure also shows the use of four rings instead of three, the rings being a little narrower than in fig. 120.

These drawings give about the usual proportions adopted for these pistons, but there is little agreement amongst the makers. But both drawings are those of actual pistons which have given good results in practice.

Gudgeon Pin Fastenings.—The gudgeon pin is generally carried on bosses cast on the inside of the piston, as shown. In order to secure it in place it is often fastened by having the end split, as shown in fig. 124, a

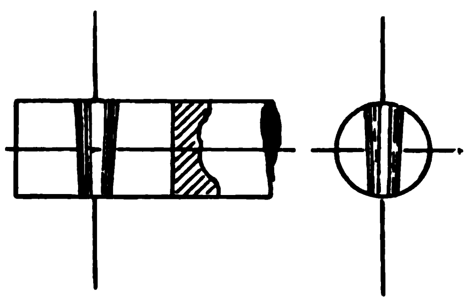


Fig. 124.

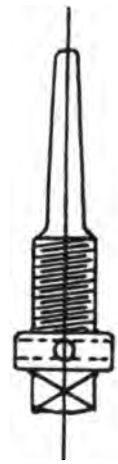


Fig. 125.

taper hole being drilled through the split. Into this a taper screw (fig. 125) is screwed; this presses the pin against the sides of the hole and makes a very secure fastening, *provided everything fits accurately*. If any part does not fit well the pin is liable to work loose. A split pin is often put into the head of the screw to prevent it from turning or becoming loose. A better plan of locking is shown in fig. 126. In this a stiff piece of steel wire is threaded through both screws and turned over at the end. Accuracy makes the best fastening. A carefully fitted plain set screw put through the gudgeon pin also works well. Another plan is to have a broad ring like a piston ring sprung over the ends of the gudgeon pin, as in fig. 127. This does not prevent the pin from turning; but if the end of it is slotted and a narrow ring, as in fig. 128, used it will be locked.

Sometimes the pin is carried on a forging bolted to the flat top (fig. 129).

This allows the whole of the inside of the piston to be turned and hence to be rather lighter than if bosses are used; but the forging will be heavier than the bosses and more expensive.

A variation is shown in fig. 130, which allows of the pin and connecting-rod being made in one piece. This does not seem to be a good plan as both

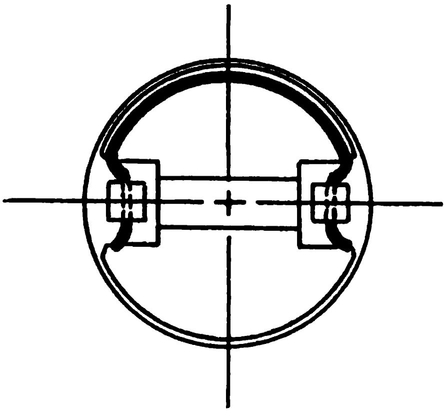


Fig. 126.

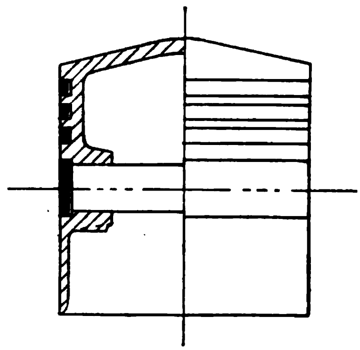


Fig. 127.

connecting-rod and pin have to be replaced should the latter be damaged or worn out. Both these fastenings are liable to leak, so that bosses seem preferable. They also bring the centre of the connecting-rod very near the top of the piston; therefore the pressure from the thrust of the connecting-rod comes almost entirely on the part where the rings are, and, therefore,

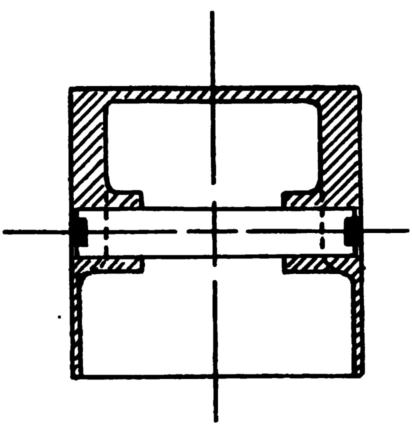


Fig. 128.

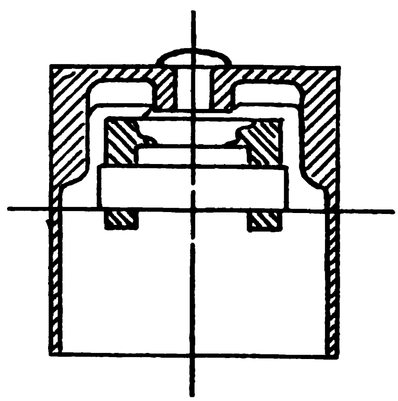


Fig. 129.

there is less surface to take it. They make the engine shorter, however, and save a little weight.

Distinctly, the cheapest of these plans is that shown in fig. 127, while fig. 128 is but little more expensive, the only difference being that in the latter the gudgeon pin has to be slotted across the ends. Both these

do away with all the set screws, and the necessary machining for them, of fig. 120, or the forgings necessary in fig. 129.

The dimensional proportions of pistons 4 inches and upwards in diameter are, roughly, as follows :—

Width of rings	$\frac{1}{20}$ to $\frac{1}{12}$	of the diameter of piston.
Thickness of rings	$\frac{1}{20}$,, $\frac{1}{15}$	,, ,,
,, top of piston . . .	$\frac{1}{30}$	,, ,,
,, piston behind rings .	$\frac{1}{20}$	,, ,,
,, piston below gudgeon pin	$\frac{1}{60}$	,, ,,
Diameter of gudgeon pin . . .	$\frac{1}{8}$	There is great diversity of practice here, some makers use a hollow pin of much larger proportion, but there is no apparent advantage.
Length of bearing in gudgeon pin .	$\frac{3}{8}$	

Very small pistons cannot be made quite as thin as these proportions would require.

The grooves in the piston for the rings must be just wide and deep enough to make a loose fit.

The gudgeon pin is made as small as possible consistent with there being enough wearing surface; the object is to keep it light. For good running it is absolutely necessary that it should be of steel, case-hardened, and ground after hardening. In this case, although very small for the pressure it takes, it usually wears better than any of the other engine bearings. Common practice is to make the diameter of the pin about one-eighth that of the cylinder, and to have the length of the bearing half the diameter of the cylinder, or rather less. Assuming a mean pressure of 100 lbs. and a maximum of 300 lbs., this gives a mean pressure on the bearing of about 1,200 lbs. per square inch and a maximum of 3,600 lbs. These are very high compared with most other engineering practice, but the bearings seem to stand very well. The bush should be of good hard gun-metal or phosphor bronze. Hardened and ground steel bushes have also been used with success, and, as long as nothing goes wrong with them, they wear beautifully; but, should they get dry, they seize so badly as to do serious damage.

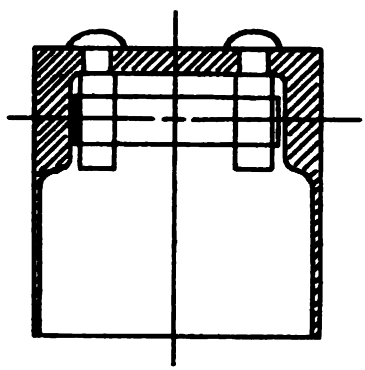


Fig. 130.

Connecting-rods.—When engines are made in quantities the connecting-rods are almost always steel stampings. Sometimes they are malleable or steel castings for cheapness, but the saving of expense is little, if anything, and the liability to breakage is greater. The greatest stress to which the rod is subject is usually the side strain due to its own momentum. For this reason it is made **H**-section, generally very deep and narrow. Fig. 131 shows a rod in practical use in an engine $4\frac{1}{4}$ inches diameter with a $4\frac{1}{4}$-inch stroke. This is probably as light as it can safely be made. It is more usual to make the section a little deeper, and this gives a greater margin of safety. Rods with a depth about a quarter of the diameter of the cylinder at the top, and rather deeper at the bottom, will be fairly in accordance with ordinary practice. The width of the section will be about half the depth at the top, and the thickness of the webs about one-eighth the depth. This

will, however, require a little modification, according to the proportion of bore to stroke in the engine; even here there is great divergence of practice with different makers. The pressures due to the working of the engine are all downwards, and from these there are no stresses on either the bottom brasses or the big end bolts. The bottom brass is often made narrower than the top for this reason, as shown in fig. 131. This should not be overdone, and the bolts should not be too small, as, in practice, the inertia of the piston puts considerable strain on them at high speeds. It is, no doubt, possible to calculate the stresses due to this, provided the maximum revolutions the driver would run his engine is known; this, however, is rather difficult. In practice the bolts are made about one-twelfth the diameter of the cylinder. In very small engines, however, they should be rather bigger than this, as they are liable to be injured in screwing up.

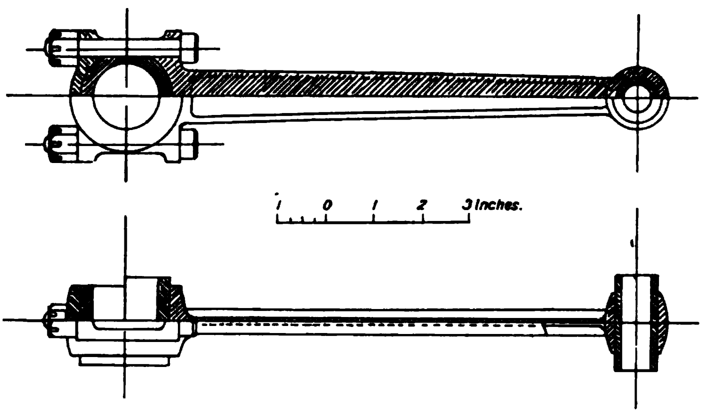


Fig. 131.

If the number required is so few that it is inconvenient to have special dies for them they may be made solid rectangular, in which case they will be heavier, as they must be nearly as deep and wide as the H-section rod. They may be also made round, and in this case can be made hollow and lighter by drilling a hole in them. In fact, probably lighter rods can be made in this way than by stamping them. In the first place, steel of a stronger kind can be used than if it has to be stamped; and, in the second, the round column is the strongest form to be had. The stamping is, however, much the cheaper for small engines. For large ones that are not made in quantities the round hollow rod is probably the best.

In some cases connecting-rods are made with the crank pin a good deal out of centre with the piston. This is not a good plan, but is done in order to have the top end pins farther apart when there is no bearing between a pair of cylinders. Fig. 132 shows the arrangement. In this case there is always a tendency for the bearings to twist, and then they do not take the load on their whole length. This may be, to a certain extent, improved if the rod is strengthened at the top and bottom ends, as shown by the dotted lines, but it is better to design the engine so that it shall be central. The length of the connecting-rods between the centres is usually from 2 to $2\frac{1}{4}$ times the stroke. The shorter the rod is, the lighter and the shorter the

engine can be made. On the other hand, for good running, fairly long rods are essential. Rods have been used less than 1¾ strokes long, but the thrust on the side of the cylinder increases very much. Fig. 133 shows the angle of the rod at its maximum for various lengths of rod from 1½ to 2½ strokes. There is little gain beyond these lengths, and the gain from a given increase of length lessens with a greater increase. Rods 2¼ strokes long seem to answer for all practical purposes.

Another point that is affected by the length of rod is the balancing. Perfect balancing is, in most systems, only obtainable with rods of infinite length, as the angle of the connecting-rod makes the half stroke of the piston non-coincident with the half revolution of the crank. In most engines with balanced cylinders, as in a four-cylinder motor, this introduces an error, as the one piston is not accelerated as fast as the other is decelerated. With a very short connecting-rod the arc of the connecting-rod very nearly coincides with that of the crank at the bottom of the stroke, whereas, at the top, it is acting in the opposite direction. The result is that the time taken to make the first part of the stroke in one direction is much greater than that in the other (see Chapter ix.).

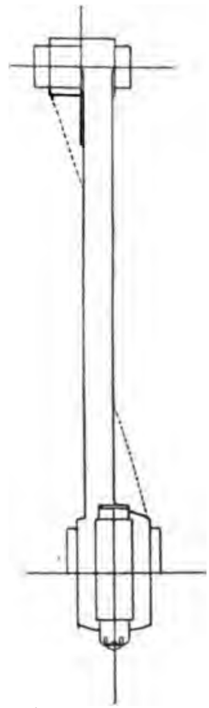


Fig. 132.

Valves.—Valves are made of mild steel, nickel steel, or pure nickel. Of these, valves of ordinary mild steel are, in my experience, perfectly satisfactory if they are large enough; they will run as much as 5,000 miles without grinding in. They are the cheapest to make and also to replace when worn out. Nickel and nickel steel do not pit so with the heat, but they are much more expensive. Cast-iron heads on steel stems also give good results, but are more expensive, as they have to be screwed together instead of being made all in one piece; they are also heavier. The wear of the exhaust valves will rather depend on the efficiency of the jacket in keeping the seating and guide cool.

The seatings of exhaust valves are almost always slightly coned, the angle being about 30°. Inlet valves, on the other hand, have either flat or

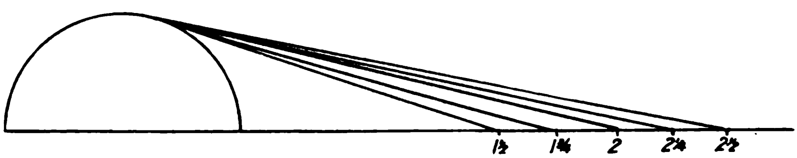


Fig. 133.

coned seatings, according to the fancy of the maker. Most of the makers of the larger class of cars began with flat-seated inlet valves, while the makers of small high-speed engines used such as were conical in form. There does not seem much to choose between them, but if the cone valve is used for the exhaust the same valve seating and valve head should be used for the inlet.

Fig. 134 shows a valve suitable for either exhaust or mechanically-worked inlet. The diameter of the stem may be from a fifth to a fourth of the clear way through the valve, but for very small valves it should be a little larger than this, as, when the size is $\frac{1}{4}$ inch or less, it is rather difficult to make them stand. The lap of the valve is about one-sixteenth to one-twelfth of its diameter. There should be a saw cut on the top of the valve (as shown) for grinding in, and the little projection is useful for rendering its removal easy with a pair of pliers.

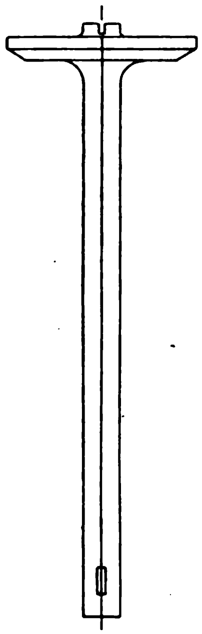


Fig. 134.

Valve Springs.—The valve is held down on its seat by a spiral spring surrounding the guide. This may be fastened to the valve in several ways. One of the simplest and oldest is to make the end of the spring hook through a slot in the valve (fig. 135). This is apt to make the hole in the valve stem rather large and is not easy to insert. Most valves now have the spring resting on a washer, which rests on a cotter, as in figs. 136 and 137. This may be better made like fig. 138, as the corners are rounded and, therefore, not so liable to break the cotter.

It might be an advantage to make the slot with rounded ends, as in fig. 139, because this can be conveniently made in a slot drilling machine, and the cotter will not be so liable to cut the valve stem.

The ends of the valve should be case-hardened, as, if not, they are liable to burr up, and then the washer cannot be removed when it is desired to take out the valve. It is a good thing to bevel off the valve, as in fig. 139, to insure that it shall not burr.

The springs that hold the valves on their seats must be strong enough to make the valve follow the cam, or else there will be a certain amount of lag and the actual motion will be different from that intended. The spring of the exhaust valve must also be strong enough to prevent the valve opening when the engine is throttled and letting the exhaust gas flow back into the cylinder. It is very easy to calculate the pressure required for this, as it is

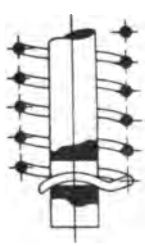


Fig. 135.

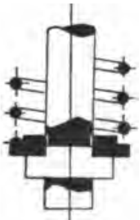


Fig. 136.

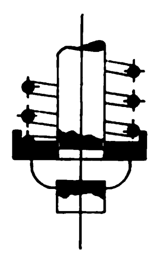


Fig. 137.

simply the area of the valve in inches multiplied by 15 lbs. Rather less than this will suffice, as there is never anything like an absolutely perfect vacuum in the cylinder.

In general, a spring of the strength given above will be strong enough to make the valve follow the cam. This will, however, depend on how fast the engine is run, the weight of the valve, and the rapidity of the opening and closing. Theoretically, if the vacuum in the cylinder is disregarded, the spring should be short, stiff, and press much harder when the valve is open than when it is shut, in order to overcome the inertia of the valve when starting to close.

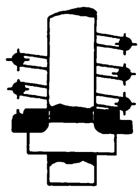


Fig. 138.

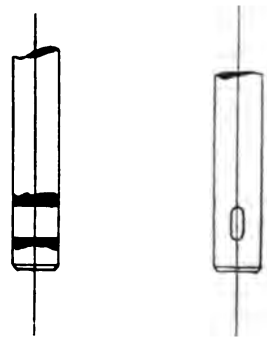


Fig. 139.

In most cases it will be best to have a valve setting in which the valve does not open and close very quickly; a spring of moderate and even strength will then be right. The inlet and exhaust valve springs should, if possible, be made interchangeable for practical reasons, and, therefore, the above remarks apply equally to them.

If the inlet valves are mechanically worked, they will have the same construction as the exhausts. In fact, they are often interchangeable, and,

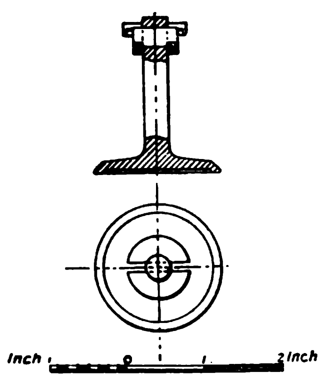


Fig. 140.

if the general design of the engine will allow this, it is in every way desirable.

If, however, they are automatic, they must be much lighter. If they are heavy, they will not open or shut rapidly enough to follow the stroke of the engine, and, therefore, the cylinder will not get a full charge of gas. This is not so evident in opening as in closing. In the latter, if the spring

that closes the valve is very weak, it will not be able to move a heavy valve quick enough to close it before the engine has completed part of its return stroke. If, on the other hand, it is strong, the valve will not open properly and the engine will not get a full charge. It is evident, in fact, that to get the best result with automatic valves a different strength of spring is required for different engine speeds. This could be arranged if it was worth while, but even then good results could not be got at high speeds with heavy inlet valves. The main point, therefore, is to make them light.

Probably no one has been so successful with the automatic valve as De Dion. Fig. 140 shows his valve for a fairway of $1\frac{1}{2}$ inches. The actual weight is $2\frac{3}{4}$ ozs., and the spring which holds it closed has a pressure of about a pound.

For the same reason that these valves must be light, they must also have a small lift, so that they may have time to close. In practice, the lift of automatic valves is about one-twelfth of their diameter.

With mechanically-worked valves, on the other hand, a larger lift can be used. In order that the area past the lip of the valve should be equal to the fairway, the valve ought to lift one-fourth of its diameter. This is not usual with motors, in which the lift is seldom more than a fifth or a sixth.

CHAPTER VII.

CRANK SHAFTS, CRANK CHAMBERS, AND CAMS.

Crank Shafts.—Crank shafts are either built up or forged in one piece. The former are more usual in single-cylinder engines, and have many advantages. In them the flywheels are usually inside the crank case, and form the webs of the crank, and the shafts and crank pin are hardened and ground, and the bearings all plain bushes and not split. When engines are built in quantities, there is little doubt that this plan is the cheapest, and, for single cylinders, the best. The work is all turned work, which can be done in an automatic lathe, and the expense of fitting split bearings is avoided. Further, the pins which take the wear are all of very simple form, and can, therefore, be thoroughly hardened and ground true after hardening. This is a very great advantage, as the wear of brasses running on a hardened ground surface is far less than what it is on a comparatively soft surface. The result is that the wear on the bushes is very small indeed, and, when the bush does get worn, the cost of renewal is less than that of adjusting a split bearing. The shafts and pins, being hardened, do not wear materially, but, when necessary, they also can be renewed very cheaply. Further, the engine with the flywheels inside runs smoother, as a rule, than an engine which has them outside, as the weight of the flywheels is in a line with the cylinder, and, therefore, there is no rocking moment in overcoming the compression. The crank chamber in this case can be split vertically on the line of the cylinders instead of being split horizontally on the line of the crank shaft; this is an advantage in many ways, as will be seen when crank cases are noticed.

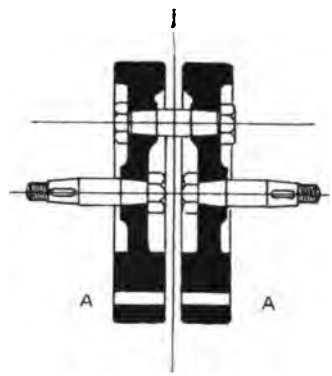


Fig. 141.

A, A, Holes bored through balance weights and reamered true for assembling crank shaft.

Fig. 141 shows the general arrangement of the crank and flywheel in a single-cylinder engine. The pins are fastened on to the two wheels with a cone and nut, there being a key also to take the twisting moment. The flywheels have weights cast on them to balance the moving parts. It is absolutely necessary that, when the crank is assembled, the two shafts should be exactly in line. Although there are keys in the ends of the crank pins, the fit of these can hardly be made accurate enough to insure this, and the parts must be tested to accuracy when assembled. There is, of course, no difficulty in getting them right in the first place in the factory, but, when they are taken to pieces by a repairer, the matter may be less easy;

but if, when they are assembled in the factory, a hole is bored through the counterbalance part of the flywheel, as shown, and reamered absolutely true, the parts are certain to be correctly placed, if, on assembling the crank again, a gauged rod is passed through this hole.

The diameter of the flywheels in this class of crank is usually made about three times the length of the stroke of the engine. The diameter of the shaft and pins is sometimes very small indeed; less than a quarter of the diameter of the cylinder. This gives very small bearing surfaces for the pressure on them, but they wear surprisingly well. The fact that the flywheels are direct on the crank pin, and, therefore, the strain of the explosion and compression is taken direct on them, instead of through a bearing, has

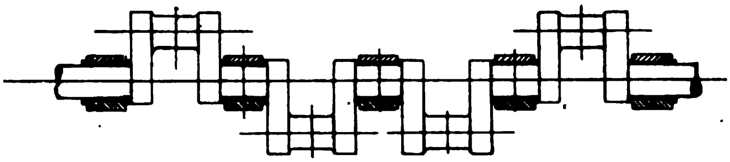


Fig. 142.

probably a great deal to do with this. Generally, the length of the crank-pin bearing is about the same as its diameter, while the main bearings are a good deal longer, say twice as long.

In multi-cylinder engines the advantages of the built-up crank are less, and the disadvantages greater. In the two-cylinder diagonal engine it is, of course, suitable, but in engines with more than two cylinders it is very seldom used; and not often with two cylinders side by side. There is no apparent reason why a very cheap and satisfactory two-cylinder engine should not be made with a built-up crank, the two connecting-rods working on to one pin; but when several cranks are built up the number of small flywheels would be heavier than the single large one with forged crank, which is generally used. On the other hand, if the weight of the flywheels

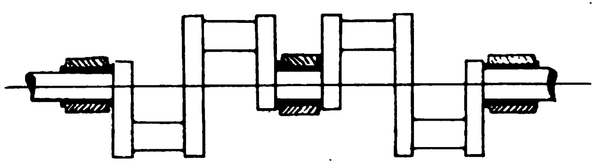


Fig. 143.

is reduced, the shafts would soon become loose for reasons which will be apparent when dealing with flywheel-fastenings.

The forged cranks are made of two general designs; in one there is a bearing between every crank, while in the other each pair of cranks has no intermediate bearing, as noticed on a previous page. Fig. 142 shows the arrangement of a crank for a four-cylinder engine with a bearing between each crank; as will be seen, it is very well supported. Fig. 143, on the other hand, shows an arrangement in which there is no intermediate bearing between the adjacent cranks, and fig. 144 shows a modification of this, in which the web is sloped instead of straight, to allow the centres of the crank to be placed further apart. The crank in this case is not nearly

so well supported, and the proportional length of the bearing is much less. When the engine is running, there is a recurrent and simultaneous downward pressure on both the adjacent cranks during every revolution, which tends to distort the crank, as shown in fig. 145. The crank is made strong enough to resist serious distortion, but there is good reason to believe that in many cases it is so distorted as to cause the bearings to wear very much more rapidly than they otherwise would do. Putting an intermediate bearing generally reduces the length between the bearings to about half, and, therefore, proportionably reduces the tendency of the shaft to spring. In order to make a smooth-running engine, it is absolutely necessary that

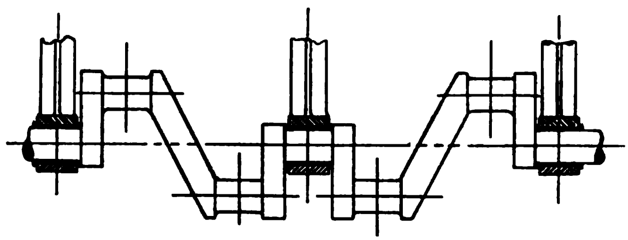


Fig. 144.

the crank shaft should be kept quite rigid. If there is the very slightest spring in the shaft, or in the bearings that carry it, there will always be a great deal of unnecessary wear on the bearings, as these will not be exactly in line with the shaft. Rigidity is, in fact, of far more importance than the amount of bearing surface allowed. For this reason, in both cases, the shaft must be much larger than is necessary to keep it from breaking, if the engine is to run well and without wear.

The only advantage there really is in the shaft with no bearing is cheapness. It has been urged that it is more easy to get three bearings in line

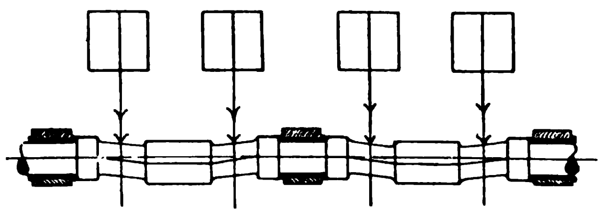


Fig. 145.

than five, but there is nothing in this, and even in the most poorly equipped shops there is no difficulty in boring the bearings in line. Even in the matter of cost there is less difference than might be expected, as the bearings are all in line in both the crank and crank case, so that no extra setting is required to machine them, and the extra cost of the two extra pairs of brasses add little to the expense.

Therefore, to have a bearing between each crank (as is usual in all other engineering practice) seems in every way preferable. If no bearing is used, every effort should be made to keep the centres of the cylinders as close together as possible, so as to reduce the distance between the bearings;

therefore, they should certainly be cast in pairs, and the walls of the cylinders should touch. The crank shaft should also be made a good deal stronger than when there is a bearing between each crank. This will, of course, make the crank more expensive, and take away some of the saving.

The size of the crank shaft may be worked out according to the usual formula, but it is not necessary to do so for the reasons given; it must be made large enough to wear well. Cranks in practice have been used from ·25 to ·45 of the diameter of the cylinder; the latter in an engine remarkable for the little wear on the bearings. When the crank is made of moderately hard mild steel, about ·33 of the diameter of the cylinder is a good working size, if there is a bearing between each crank, and ·4 if there is none. This applies to engines in which the length of stroke is about equal to the diameter of the cylinder; if the stroke is longer, the crank should be bigger.

The thickness of the webs of the crank shaft should not be less than ·6 of its diameter, and their width about 1·25; it is better to make the webs stronger. If there is no intermediate bearing, the thickness of the web between the two adjacent cranks should be more than this, as the stress on it is great; it should be stronger than the shaft.

The proportioning of the length of the brasses for the crank pins and main bearings requires some consideration. In most engines the total length of these is limited, as it is desired to keep the engine light and, therefore, short. In some cases, however, the centres of the cylinders are thrown so far apart that there is no need for any limitation. Taking the crank-pin bearings first, if no restriction is needed, the best length is probably about 1·25 times their diameter. In some cases they have been made a good deal longer, in order to get more wearing surface, but the advantage is doubtful, as the tendency of the crank to spring does more harm than the extra surface does good. In any case they should not be more than 1·5 diameters long. If length is limited, a crank pin with a length equal to the diameter, if well supported, would probably wear nearly as well as a longer pin. In fact, to get good working results it is better to increase the diameter rather than the length. If there is ample room, the space not occupied by the crank pins may be occupied by the main bearings; but if space is limited, the main bearings should not be made shorter than the crank pins, and, if anything, it is better to make them a little longer; and it would be advantageous to make the main bearings and crank-pin bearings interchangeable, if the design admits of this. It is, however, generally found that main bearings require more surface than crank pins to make them run equally well, and, further, it is generally much easier to adjust the crank pins than the main bearings.

The corner between the crank web and the shaft and pin should be rounded; if sharp, cracks are liable to start in it.

The crank shaft and pins are almost always made of the same diameter, though, in theory, the crank pin might be made the larger of the two. The flywheel is generally fastened on, either with a flange (as shown in fig. 146), or by a taper and nut (as shown in fig. 147). The former plan is almost universal in engines with a forged crank, while the latter is usual with the built-up ones. If the crank is forged there is no inside flywheel; hence the fastening of the flywheel is subject to a reversal of stress every working and every compression stroke, and, therefore, a cone and nut seems liable to work loose, though there is no doubt that it can be made a satisfactory

fastening if the cone is long enough and the taper is carefully fitted. The flange is, however, absolutely secure, and, therefore, it is generally preferred, notwithstanding that it is more expensive. On the other hand, the taper and nut seem to be a perfectly satisfactory fastening for a flywheel in an engine with inside flywheels, as the latter take the engine over the centres, while the outside flywheel serves to carry the clutch rather than to act as

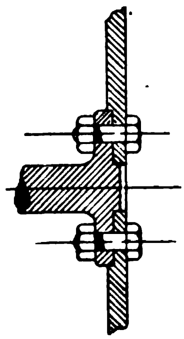


Fig. 146.

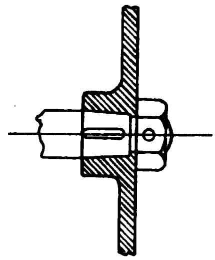


Fig. 147.

a flywheel, except, perhaps, as a reinforcer when the engine is running very slowly.

Material.—The crank shaft should be made of the hardest steel that can be economically worked. Experience is that the wear in the brasses supporting the shaft depends very largely on the hardness of the shaft itself. The hard steel soon acquires a surface with a glossy polish which has very little wearing effect on the brasses, whereas the soft steel never acquires this surface, and, as a consequence, it wears the brasses more rapidly than the polished harder steel does.

If there is no flange, or only a small one at the after end of the shaft, probably the cheapest way of making the shaft is to saw it out of a slab with a band saw. If, however, the flange at the after end is large, the best way probably would be to turn the whole out of the solid; in this case there is no reason why the crank webs also should not be turned. Fig. 148

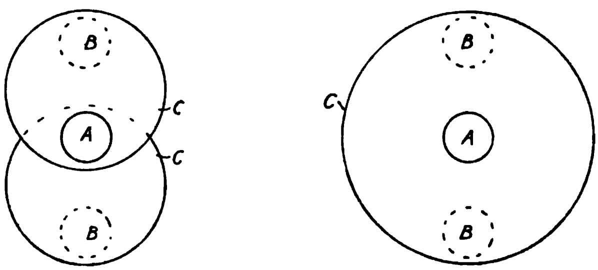


Fig. 148.

A, Crank shaft. B B, Crank pins. C, Webs.

In the one case the shaft has to be set on different centres to machine the webs. In the other these are machined at the same setting as the crank shaft, which is, therefore, slightly cheaper though heavier.

shows two ways of doing this. In the second the crank web is turned on the same centre as the main bearings, and, therefore, at the same setting; this is cheap. Both plans make very strong crank webs, the web being very wide where the stress comes on it, and the crank shaft is, therefore, rigid.

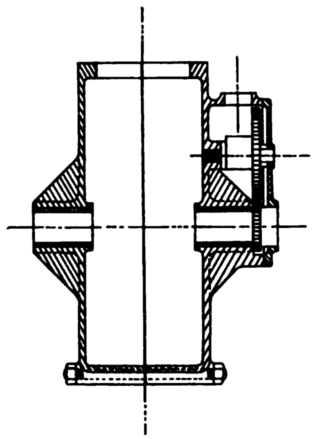


Fig. 149.

Crank Chambers.—The simplest crank chamber is naturally that for a single cylinder engine. This can be split either along the line of the crank shaft or vertically along the line of the cylinder. In the former case the construction is similar to that of an engine with several cylinders and will, therefore, not require special consideration. The latter is, however, the most usual.

Fig. 149 shows the general arrangement of such a crank chamber. It is split on the line of the cylinder and the joint is a turned one, being faced at the same operation as the boring of the seatings for the brasses. The bearings are not split, the cranks being built up and the pins hardened and ground; when the bearings wear new bushes are put in.

The valve gear in this type of crank case can be arranged in several ways. As a rule, the cams and runners are outside the crank case, as in fig. 150, and at one end. In this case, if mechanical inlet valves are used, both may be placed on the same cam shaft, as in fig. 151, or on two cam shafts, as in fig. 152. The former

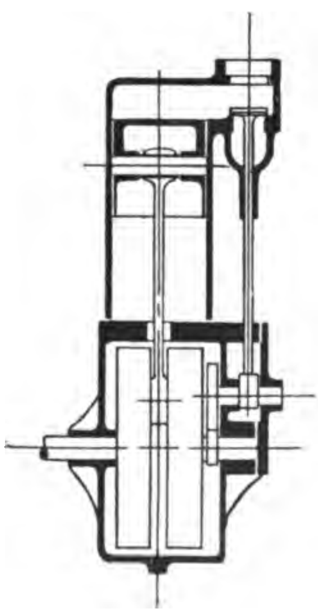


Fig. 150.

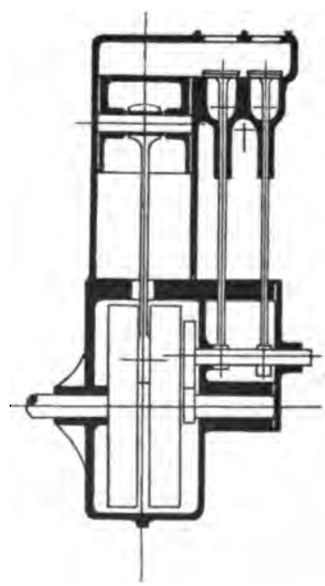


Fig. 151.

is the cheapest to make, but the engine has to be longer; a further objection is that the same pattern cylinder will not be available for engines with more than one cylinder. Occasionally the cam shaft is arranged at the side of the cylinder, as in multi-cylinder engines, and

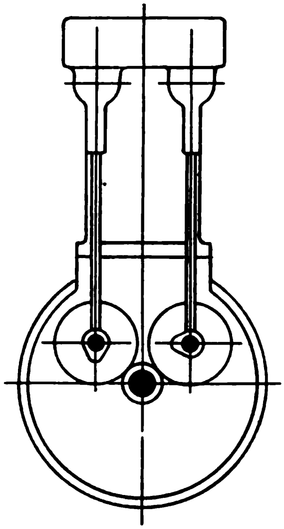


Fig. 152.

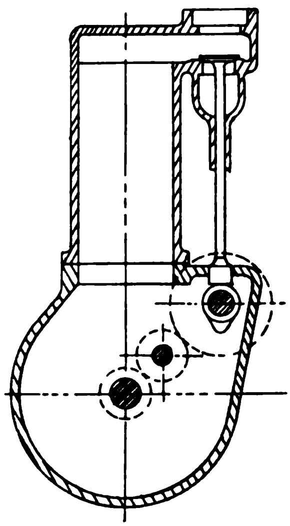


Fig. 153.

then both valves may be on one shaft with the ordinary arrangement of cylinder. The cam shaft has, however, to be carried out further from the crank shaft than in multi-cylinder engines in order to clear the flywheels

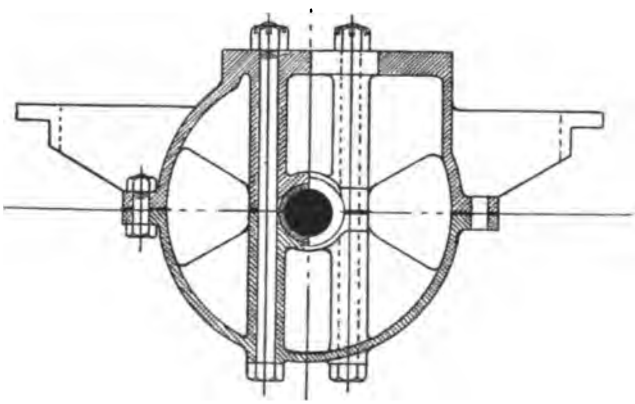


Fig. 154.

(fig. 153). For single-cylinder engines the crank case made in this way seems better than that split in the line of the crank shaft, for the following reasons:—It is cheaper, as the joints are faced at the same time that the bearings are bored. Also, it is easier to make the turned joint on the crank

chamber oil-tight than the one on the line of the crank shaft. The hardened and ground pins wear very well in the solid bushes, and it is nearly as cheap to renew these when ground as to adjust split bushes. There is no reason why this construction should not be adopted for two-cylinder engines of small size, although it has seldom been used. The cylinders would, of course, have to be cast in one and the shaft run in solid bushes in the same way. This would make a very cheap and suitable engine for small cars.

If the valve gear with these crank cases is at the end of the shaft it can be arranged in two ways. The gear wheels can be inside the crank case, as in fig. 150; or outside, in which case the cam shaft and gear wheel can be made in one piece and be carried on a stud, if desired, as in fig. 149, which makes a cheap arrangement.

For all larger engines and those with more than two cylinders the crank case is usually split on the line of the crank shaft. Fig. 154 shows the older form of construction. In this the bearings are held between the halves of the crank case, a constructive defect, as it is not possible to take off

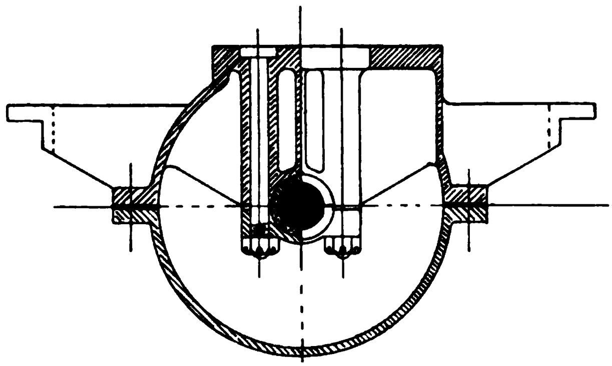


Fig. 155.

the bottom half of the crank case without dropping out the crank shaft, while the adjustment of the bearings, so as to fill the space between the halves of the case, is very difficult to manage. In fact, an engine of this kind can only be repaired in the shop.

Fig. 155 shows an improved arrangement which is coming greatly into use. In it the crank shaft is carried entirely on the top half of the crank case and the bottom half is formed by an oil cover. In fact, the engine can be run without the bottom half on. As the bottom half is quickly removable all the parts can be examined in place and the bearings adjusted without taking the engine out of the car. It is also easier to machine, as when boring the seatings for the bearings the work can be examined.

Another arrangement is shown in fig. 156. In this the whole of the engine is carried on the bottom half of the crank case, which is the one fixed to the frame. This is not so accessible in a car as the last plan, as to get at the crank all the cylinders have to be taken off and their pipes disconnected. It is better than the plan shown in fig. 154, however. It is very suitable for boat work if large doors are provided for easy access to the crank, an easy matter in marine engines, as these are larger than those in cars. The great convenience in car construction is that the crank case

and gear box can conveniently be made all in one casting, but this hardly compensates for the difficulty of getting at the parts.

Another plan is that shown in fig. 157, in which the top half of the crank case is fastened on to the frame, while the main bearings are carried on the bottom half. In this plan the whole of the inside of the engine can be withdrawn through the bottom of the crank case. This is not nearly so

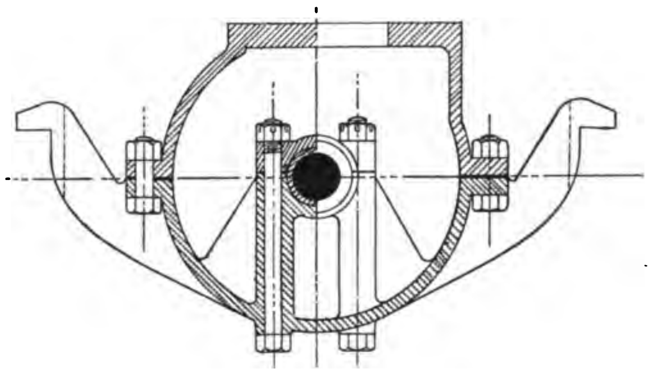


Fig. 156.

convenient as the arrangement shown in fig. 155, as it does not provide for inspection of the parts in place.

Inspection doors are often provided in the top half of the crank case, but are generally too small to be of much use, and are a source of expense. They are quite unnecessary if the bottom half of the case can be taken off easily.

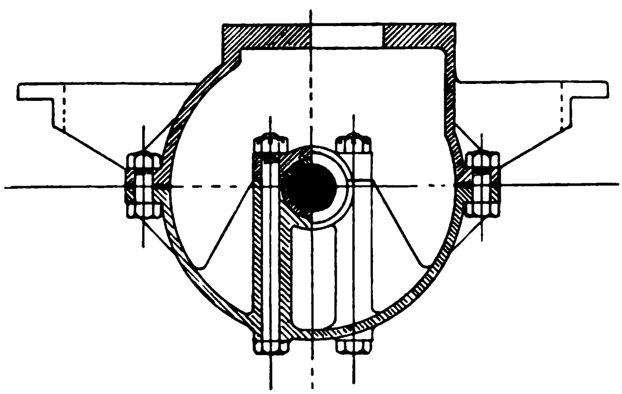


Fig. 157.

Cam Shafts.—The arrangement of the cam shaft depends on that of the valves. If the valves are opposite there must be two shafts, but the arrangement is the same as when the valves are on the same side.

In the older engines the cams and shafts were often entirely unenclosed. As they were open to the dust the wear must have been greater. They are now generally enclosed.

The cam-shaft bearings can be arranged in several ways. Fig. 158 shows a cheap way, but the cams have to be fixed to the shaft after the latter has been threaded through the bearings, and there is a liability of their not being so well fixed as when this can be done from outside.

Fig. 159 shows the cam shaft secured by what is practically a loose cap the whole length of the crank case. This is a good plan in many ways, but is rather expensive and heavy, as it entails a flange joint the whole length

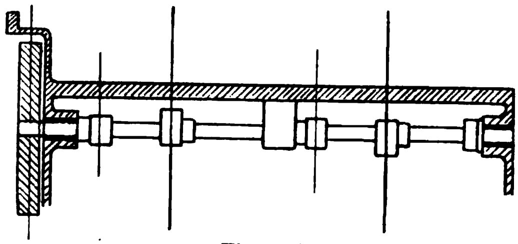


Fig. 185.

of the crank case. It is generally at such an angle as to require special machining, and requires a good many studs to hold it on.

If the cam shaft is reasonably short the cheapest way is not to have any intermediate bearing, but to make the shaft strong enough to run with only a bearing at each end, as in fig. 160. If the shaft is too long for this, a bearing can be put in the middle by making the outside diameter of the brass large enough for the cams to thread the seating, as in fig. 161.

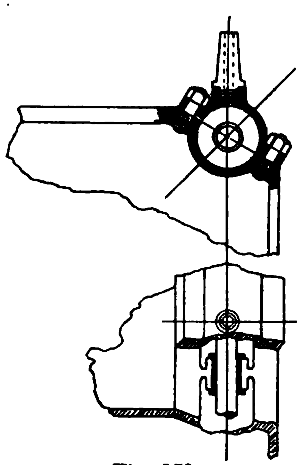


Fig. 159.

Fig. 162 shows another plan, which seems cheap and good. The bearings are all bolted up to seatings at the top of the crank case, and these can be machined at the same setting as the main bearing seatings.

In some engines the gear-wheels driving the cam shafts are all outside the crank case. There seems to be no advantage in this, which is probably a relic of the days when the cam shaft was outside. Sometimes they are enclosed by bolting a casing up to the main part of the crank case, as in fig. 163. This entails an extra joint as compared with the plan of simply putting them in through the bottom of the top half, and is, therefore, more expensive and heavier. If the whole of the engine is carried on the bottom half of the crank case, as in marine engines, it is a good plan as the gear-wheels are easily accessible from the top.

Fig. 164 shows a general arrangement of a crank case suitable for a four-cylinder engine. It is for a four-cylinder engine with the arrangement of gear-wheels shown in fig. 30. There is a bearing between each crank, and the cam shaft is carried, as in fig. 160, without an intermediate bearing. The projection at the front end is for the governor, while the seatings for the pump and magneto are on the front arms. The top of the crank case is made fairly thick and the bearings

are well webbed up to it to make them thoroughly rigid, as the greater the rigidity the less is the wear on the brasses. The bottom half of the

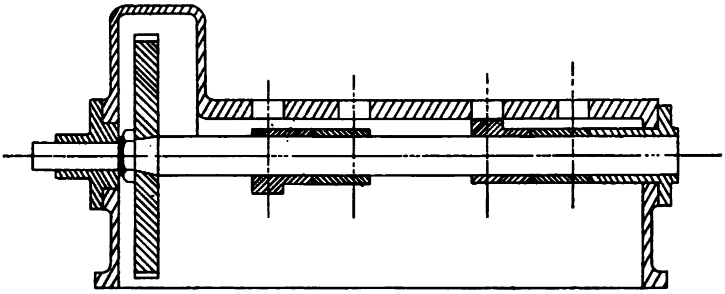


Fig. 160.

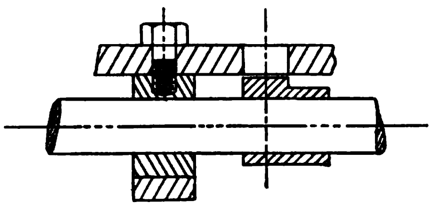


Fig. 161.

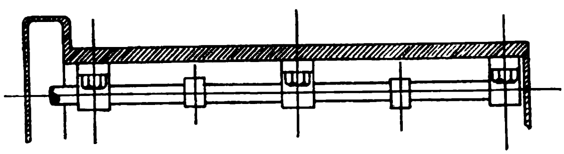


Fig. 162.

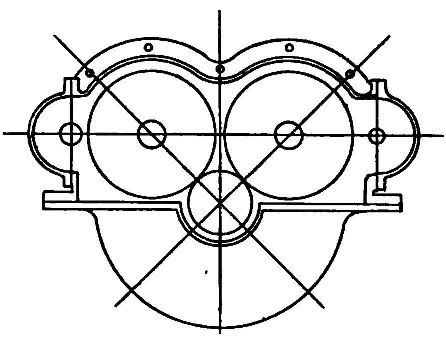


Fig. 163.

crank case is only an oil cover, which can be taken right off without disturbing any of the working parts.

In designing crank cases care should be taken to arrange for all the

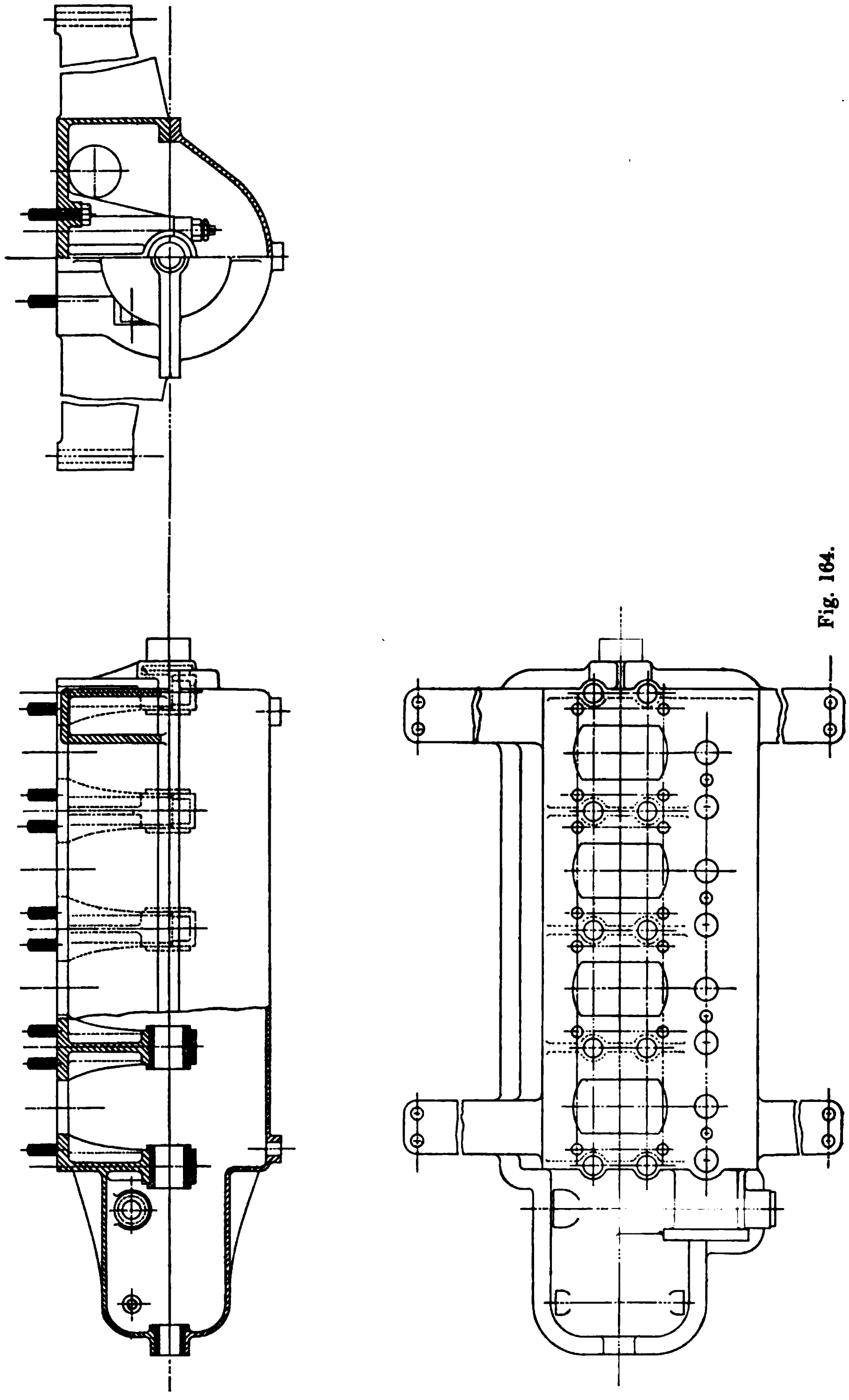


Fig. 164.

seatings, &c., that may be required to be machined at as few settings as possible. This applies also to all connections to the case for oil, &c. Often oil connections, &c., are put on at odd angles necessitating a special setting to machine, when they could just as well be set vertical or horizontal, and arranged to be machined at the same setting as other parts.

If the carburettor is carried on the crank case there should be a machined seating on both. It is much too common to see carburettors, &c., carried on bits of bent iron fastened to the arms of the crank cases, &c.

Crank cases are generally made of aluminium, which is doubtless the best material for pleasure cars. In most cases the greater part of the case is made as thin as it can be satisfactorily cast, the bearings, &c., subject to great strains being made more substantial. If the bolts reach to the top of the crank case, as in fig. 164, the latter is not liable to tensile strains, as those are taken by the bolts, the heads of which are quite close to the cylinder bolts. The bottom half of the case is also made of aluminium, though the lightest that could be made would probably be composed of sheet brass, soldered and rivetted to a rim to make the joint. This has sometimes been used. Cast iron is best for ordinary marine work, and probably for commercial work, as it makes a very rigid crank case, and, there fore, good wearing bearings.

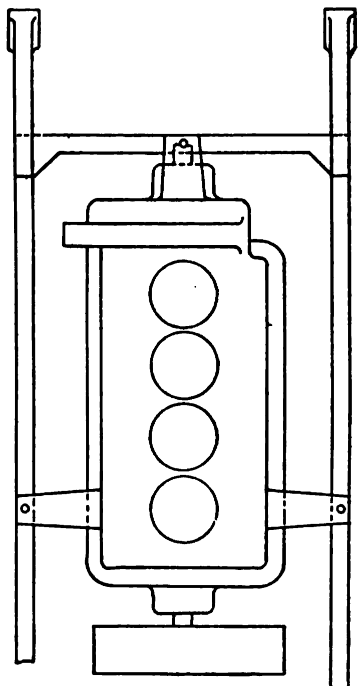


Fig. 165.

Both the main bearing bolts and those that hold the cylinder down must be strong enough to take the stress of the explosion, the sizes for which are given in Chapter vi. As the bearing bolts have to take the same stress as the cylinder bolts, they may be of the same size.

When crank chambers are of aluminium, studs screwed into them should be avoided as they are apt to work loose, and, in particular, the cylinder and main bearings should be held in place by bolts, as shown, and not studs.

The crank case must have arms cast on it to support it on the frame, and the design of these will depend very largely on the arrangement of the latter. If there is no inside frame the arms must be long enough to stretch right across the main frame and have the necessary rigidity. Sometimes the casing that encloses the gear wheels driving the cam shafts, &c., can be used to support one end of the crank case; this makes a very neat arrangement. Occasionally arms stretch across the frame at both ends of the crank case; but it is generally lighter to carry the front end on the cross member of the frame, as in fig. 165. If this is done the long arms at the front end of the crank case are saved, and three fixing flanges and bolts,

&c., will suffice instead of four, thereby saving a little both in weight and cost.

It is claimed that this "three-point suspension," as it is called, allows of the frame of the car springing without twisting the engine bedplate; but the frame that does not spring seems preferable.

If there are arms at the fore end of the crank case as well as at the back, it would seem that the cross member of the frame at the front end might often be dispensed with. Possibly this would be both cheaper and lighter than having a cross member built into the frame and then bolting the crank case to it.

There is no doubt that an underneath covering which excludes the dust, like that shown in fig. 166, keeps the engine cleaner, but it adds more to the weight of the car, intensifies the car noises to the riders in it, and makes the machinery less accessible. If the casing is often removed it gets bent out of shape, and is then difficult to replace. Again, this underneath casing is a very inconvenient receptacle for accidentally dropped small articles like nuts and screws, as they are difficult to find. Groping under the engine is filthy work, and if the casing has to be taken off it may be the work of hours.

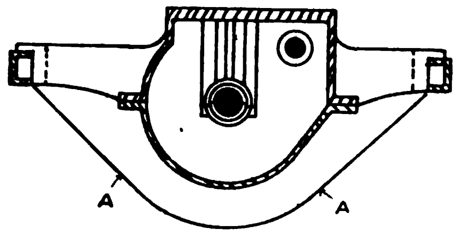


Fig. 166.—A, A, Casing under engine.

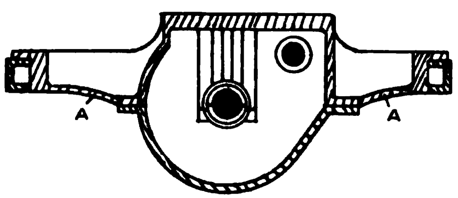


Fig. 167.—A, A, Casing under engine.

As a matter of fact, the advantage of casing in the engine underneath is much more apparent than real. The working parts of the engine should be so encased in the crank chamber that dust and mud will not get into them. The dust that does the most harm to the engine is that from a dry flint road. With the ordinary arrangement of radiator and bonnet this has access to the engine at all times, and, in fact, there is generally a fan which blows the dust thereon.

A casing can, however, be arranged which does away with most of the disadvantages of the ordinary form, and yet thoroughly protects the engine underneath. This is done by making the casing as in fig. 167, so that it is on a level with the split in the crank case and does not go right underneath it, but only up to it. A great deal of weight is thus saved, as the part under the crank chamber is dispensed with. Further, one can take off the bottom of the crank case without disturbing the casing, and also can reach anything that may be dropped into it without groping under the engine. Being much more rigid it also does not increase the noise so much.

The casing is sometimes cast on the crank chamber, so that it practically forms a continuous arm for the support of the crank case with vertical webs at each end to stiffen it. This is neat, probably little heavier than one made of sheet rivetted on, and certainly cheaper.

Cams.—Cams and the runners, or rollers, which run on them are almost

invariably made of steel case-hardened. Their shape must naturally be made for the valve setting that is required.

Figs. 168 and 169 show the shape of two cams and runners; the one gives very quick opening to the valves and the other a slower one. These figures also show the relative valve settings in terms of the stroke of the engine. It will be seen that the difference in the valve opening at different parts of the stroke is not so marked as might be expected, and when the piston is travelling at its fastest on the exhaust stroke the slow opening cam

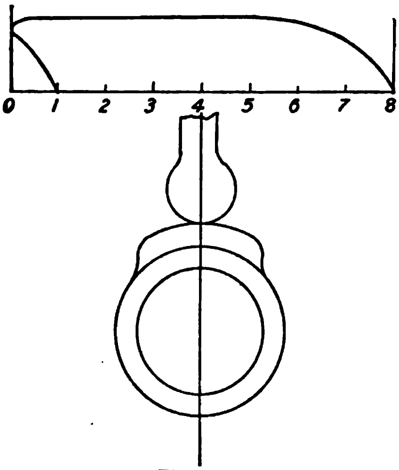


Fig. 168.

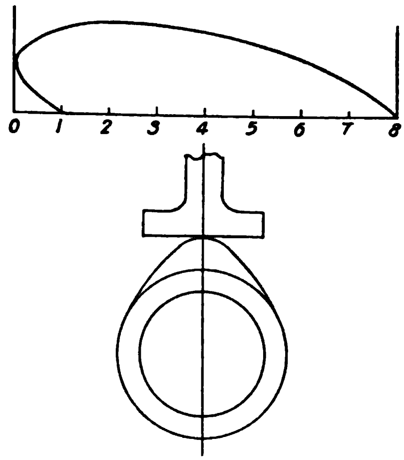


Fig. 169.

This valve is given more lift than is possible with a quick opening cam as in fig. 168; hence its greatest opening is greater.

has the advantage. The cam with the slow opening and closing will, generally speaking, give much the quietest running as also the least wear and tear, and should the valve opening not be large enough with it at the beginning and end of the stroke, it would probably be better to make the valve larger than to make the cam open and close more quickly.

Cams are usually made loose from the shaft they are on and are often fixed with a taper pin like that shown in fig. 170. This is quite a good fastening for small engines, if well done. Probably a better fastening is to have a key, as it would then be easier to make them to gauge.

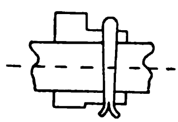


Fig. 170.

Cams are sometimes made solid with the shaft and machined out of one piece. This has the advantage that they cannot possibly come loose, but is expensive, and generally entails the bearings which carry them being split. The great objection seems to be that if anything goes wrong with *one* of the cams the whole cam shaft and all the cams have to be renewed.

The runners which take the motion from the cams are sometimes made with rollers and sometimes without. If the cams are all enclosed in the crank-chamber, and, therefore, well lubricated, there seems to be no advantage in the roller, which adds to the expense, since a plain runner with ample surface will wear perfectly.

Various arrangements of runners and the guides they work in are shown in figs. 171 to 175. It will be seen that there is an enormous difference in the amount of work in them.

The cheapest form of runner is that shown in fig. 171, which is round, and can, therefore, be finished in a turret lathe at one operation. This, with such a cam as that shown in fig. 169, works perfectly well in practice.

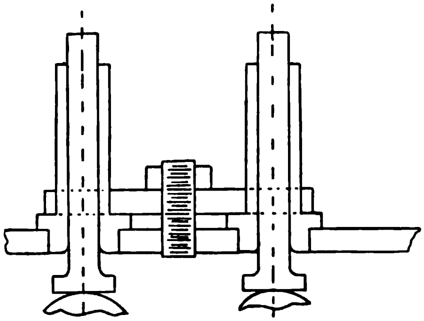


Fig. 171.

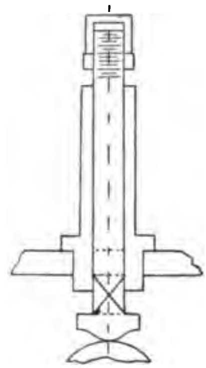


Fig. 172.

As the head is flat there is no need to have any arrangement for keeping it from turning round. The guide may also be round, and in this case can also be finished at one operation. Generally the cheapest way of fixing these in the crank case will be as shown, each pair being held by a dog with one bolt to secure it.

If a quicker opening to the valve is desired the runner must either have a roller or be suitably shaped to open quickly, as indicated in fig. 172; but this must be prevented from turning round, either by being placed in a square guide or by means of a set screw, as in fig. 173. In the latter case, the guide must be prevented from turning; in fig. 173 the guide is held on by two studs. In any case, the runner and guide will cost two or three times as much as the plain rounded one represented in fig. 172.

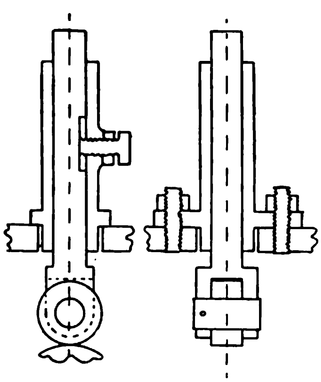


Fig. 173.

If rollers are used they are arranged somewhat as in figs. 173, 174, 175. Of these, fig. 175 seems the cheapest; it is very neat, as the guide and runner only require to be turned and slotted and the pinhole drilled; the guide keeps the pin in place. In fig. 174 the runner is kept from turning by a fork that fits on the shaft; or a double fork may be used, or the roller may be held on by an overhung pin.

If the runner does not need to be prevented from turning by the guide, this may be screwed into the crank case, as in fig. 174; but the plan is not

good if the part it screws into is aluminium or cast iron, as, if the thread is damaged, it is difficult to repair satisfactorily.

Sometimes the runners are adjustable in length, as in fig. 172. This adds slightly to the expense, but is very convenient, since it is always possible to adjust them so as to have the smallest possible amount of clearance; this makes them run quiet. It also insures that the valve has a full lift. If the parts are made absolutely to gauge it is not really necessary, but is certainly convenient. In setting the height of this it must be remembered that the exhaust valve expands with the heat when the engine is running.

The cams and runners for the low-tension ignition will follow the general arrangements of the valve cams, &c., but will, of course, have to be suitably shaped and arranged for the special kind of ignition.

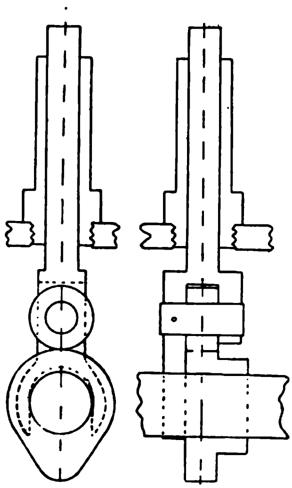


Fig. 174.

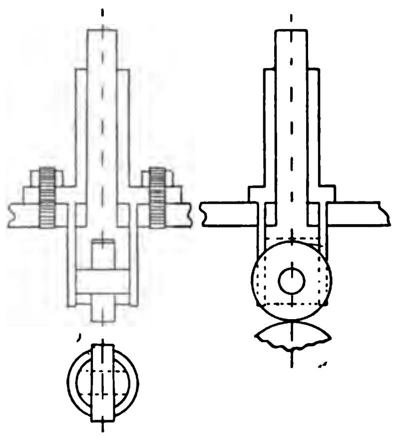


Fig. 175.

In order to wear well cams are generally made with their width about a third of the diameter of the valve, or a little wider would be still better. They should not be made too small in diameter, as, although this decreases the rubbing speed, it makes the curves sharper and, therefore, increases the crushing load. The rollers, with which these are used, should also be of good diameter, for the same reason.

The diameter of the shaft depends on the distance between the bearings, size of valves, and the rapidity with which they open; it is hence impossible to give rules for it, but in practice it is made anything from a third to two-thirds the diameter of the valve. It is not well to cut this too fine as the strain on opening a valve rapidly at high speed is considerable.

CHAPTER VIII.

PUMPS, FLYWHEELS, AND PIPE ARRANGEMENTS.

Pumps.—The water circulation in cars is generally effected by a pump. Reciprocating pumps were general in the early cars, but rotary pumps are now almost universal. Of these the simplest form is the centrifugal, shown in fig. 176, and for cars the smallest efficient kind is preferred, as economy of power is far less important in them than in engineering work generally, in which economical working is of primary importance. In consequence of this difference the motor pump has the case made as small in proportion to the size of the runner as it can be, whereas in other machinery the casing is very much bigger than this.

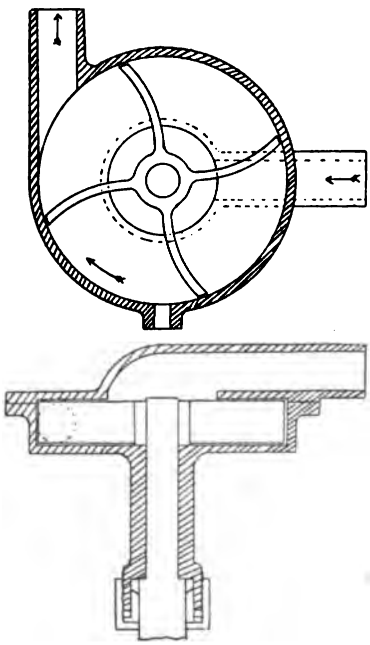


Fig. 176.

It is certainly the cheapest to make and the most convenient to use, as there is only one moving part in it, and this need not be water-tight. For the same reason it is not liable to get out of order, and should any dirt get into the circulating water this is not liable to cause excessive wear.

The older way of driving this pump was to have a small friction wheel running against the flywheel of the engine. This is rather complicated, as a spring is required for keeping the friction wheel in position, and either the pump must be mounted so that it can move slightly to accommodate this, or the shaft must have a flexible joint in it. These additional requirements add slightly to the expense, more especially in erection. Further, the friction wheel and its bearings sometimes give trouble.

The advantage of this method of driving is that as the pump speed can much exceed that of the engine, the pump can be made smaller than if it were less. Still the weight of the friction wheel, bearings, and brackets equals the amount thus saved.

In most modern cars these disadvantages are avoided by mounting the pump on the engine and driving it by gear, very often on the same shaft as some other part of the gear, such as the magneto. In this case a larger pump must be employed, but as it does not run so fast it does not give so much trouble. The size varies much, as it mainly depends on

the kind and size of radiator and pipe arrangement used in the car. If the radiator has a fair way through, and the pipes are simply arranged, the pump need not be so powerful as when the conditions are reversed.

With a radiator of ordinary construction many makers adopt gear-driven pumps of such a size that the speed of the tips of the blades is about 1,000 feet a minute when the engine is running at normal speed. Friction-driven pumps generally run a good deal faster than this.

Fig. 177 shows a pump very commonly used, which, if well made and designed, gives good results. It does not seem, however, to have any

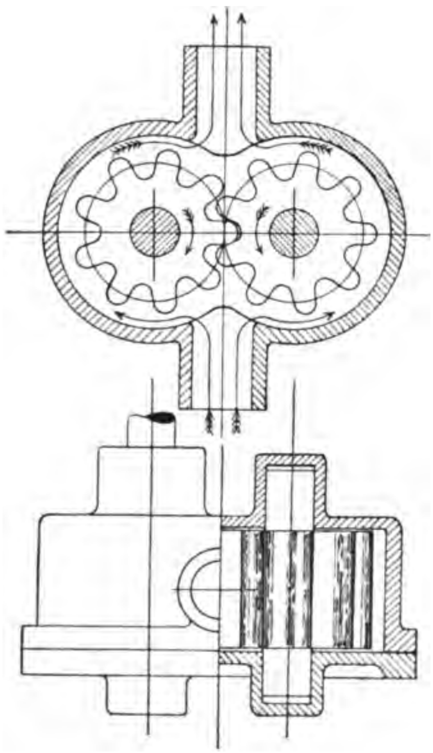


Fig. 177.

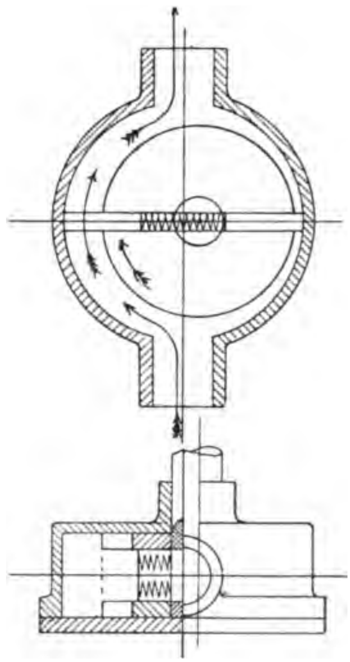


Fig. 178.

advantage over the centrifugal, and it is much more costly to make. The casing has to be machined on two centres instead of one, and the spindles have four bearings instead of one, while the wheels must have teeth cut in them. Like all positive pumps, it is much more liable to wear from dirt in the water than the centrifugal, and should any wear take place the wheels cease to be a tight fit in the casing and the pump does not work.

Fig. 178 shows another form of positive pump, in which the spindle is eccentric to the casing, and has a sliding shutter in it. This is cheaper than the last, but the shutter seems very liable to wear.

Fig. 179 shows a modification of this, which is better, as the friction of the shutter against the side of the casing is avoided.

Fig. 180 shows a modification of fig. 178, in it rollers replace the shutters; it is cheaper to make and less liable to wear.

In some pumps both the spindle and the bearing in which it runs are made of brass; this occasionally causes great wear. It is true that they will run all right as long as they are lubricated, but this entails a great deal of grease being used as the water gets hot when the engine is running, and, consequently, the grease melts and flows into the water, thereby reducing the efficiency of the cooling surfaces.

Steel spindles on brass bearings are better, but in this case the steel is liable to rust. The best plan seems to be to have brass spindles and white metal bearings, as this will run very well with water lubrication, grease being dispensed with. This is the practice on steam launches for

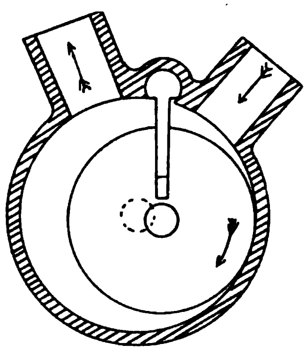


Fig. 179.

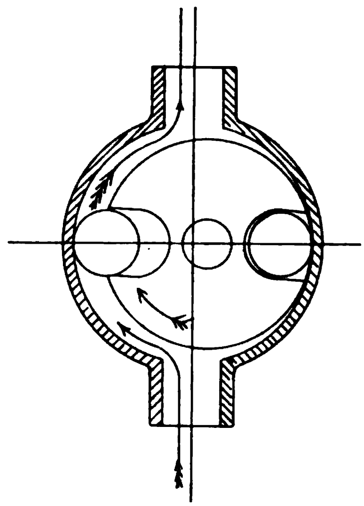


Fig. 180.

the tail ends of screw shafts which have to be run with water lubrication. Small shafts are usually run in a special white metal alloy composed of 80 per cent. zinc, 16 per cent. tin, and 4 per cent. copper. There is no reason why this alloy should not be the best for pumps; it is a very cheap one. Other white metals will run very well.

There is usually a stuffing-box to prevent leakage at the place where the spindle passes out of the pump casing, as in fig. 181. Sometimes the spindle has a very long bearing, and grease is used to keep it water-tight, but it gets very leaky as soon as the bearing wears.

A very simple plan is to drive the pump by a spring, as in fig. 182, fitting into a notch at the end of the pump spindle, and pressing against a collar with enough force to keep it tight.

Flywheels.—The flywheels of single-cylinder engines are much lighter than those with two or four cylinders. This is probably largely because the flywheel is generally enclosed in the crank case, and, therefore, to put in a

large flywheel makes the crank case very big. Still they are, in practice, big enough, and it is hence a question whether it is necessary to use as big flywheels as are usual in engines with several cylinders. At first sight it would seem that the flywheel could be reduced nearly in proportion to the number of cylinders. The flywheel of a single-cylindered engine has to take the engine through a complete revolution and one compression stroke between each explosion and the next. With two cylinders, and the working strokes even, one revolution only is needed instead of two. With a four-cylinder engine, one cylinder is compressing, and the preceding one working, while the flywheel only carries the engine over the dead point Thus, in theory, the flywheel might be proportionally much smaller in two- or four-cylindered engines, even with the same sized cylinders, but this assumes that there are no miss fires. On the other hand, should one of the cylinders in a two-cylindered engine miss fire, the flywheel has to take the engine over two compression strokes instead of one before the next working stroke.

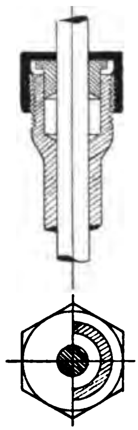


Fig. 181.

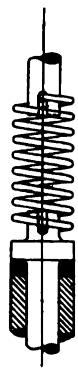


Fig. 182.

It is, therefore, not advisable to reduce the flywheel too much. Still, it may be doubted whether it is necessary to make the flywheel of a four-cylindered engine both larger in diameter and a great deal heavier than that of a single one of the same size, as is often done.

A large flywheel certainly makes the engine run steadier, as it absorbs the individual strokes of the engine, and makes the turning moment more uniform. On the other hand, when the engine is running light a very heavy flywheel vibrates the car more than a lighter one would. Those who have carefully watched a car with a cut-out governor running light will have noticed that when the governor cuts-in the whole front of the car gives a twist. This is because the flywheel will not start the instant the explosion occurs, and, therefore, the engine tends to twist the front of the car in the opposite direction. The same thing occurs at every stroke with an engine running light and slowly; hence the heavy flywheel vibrates the car more than a lighter one.

As the flywheel is often about one-third the weight of the whole engine, it is important to keep it as light as possible. In an engine with low com-

pression it will be possible to make it lighter than in one with high compression. To make a flywheel as light as possible, it should be large in diameter, and as much weight as possible should be in the rim. In fact, we can reduce the weight of a flywheel in direct proportion to its diameter, and still have the same momentum in it. The diameter will generally be determined by considerations of space.

The design of the wheel itself is generally made to suit the clutch and crank shaft. Many flywheels are made with a plain disc and rim, but this is wrong in principle, as an undue proportion of the weight is near the centre. Often they are now made with spokes like those of an air-propeller. In many cars this is the only fan used to keep the air moving through the

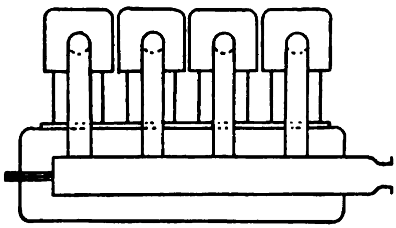


Fig. 183.

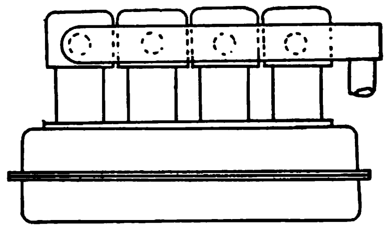


Fig. 184.

radiator, but, even if this is not so, it is a good form of wheel, as it throws a current of air on to the clutch.

If the engine is single-cylindered, and the flywheels are inside the crank case, their diameter is generally made about three times that of the piston, and their weight about 1 to 1½ lbs. per cubic inch of cylinder capacity.

If they are outside, they are often a good deal larger in diameter, and twice as heavy, even for multi-cylinder engines, but there does not seem sufficient reason for this. Flywheels are, practically, always made of cast iron.

Pipe Arrangements.—The pipe arrangement of an engine is a matter of great importance. All expensive pipe-work should be avoided, and the

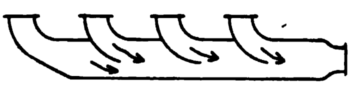


Fig. 185.

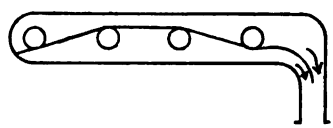


Fig. 186.

arrangement should be such as will give rise to the smallest amount of inconvenience.

For instance, separate pipes are often taken from each cylinder to a common exhaust pipe, such as is shown in fig. 183. In this case, to get at the exhaust valves is very difficult, and, if the pipes are hot, practically impossible. A better plan is to have a casting to take all the exhaust outlets, and one pipe to the silencer, as in fig. 184. All the valves are then accessible, and the pipe well out of the way. It is sometimes said that this may disturb the exhaust, as in each pair of cylinders one valve is opening just before the other shuts; consequently, there is a sudden puff of exhaust

in the pipe just before the valve shuts. This may pass from one cylinder into the other just before the valve closes, and into the carburettor when the inlet opens. This will not happen if the pipes are big enough, but it can be avoided either by making the branches, by which each exhaust joins the main pipe, of the shape shown in fig. 185, so that the action of each puff will have a tendency to draw out the gas from the other cylinders; or, by having a partition down the box, as in fig. 186, to keep the two streams separate till they are too far from the cylinder to do any harm. It is found in practice that, if kept separate a little way from the cylinders, they have no appreciable effect on each other, and the plan, common years ago, of having separate pipes for each cylinder the whole way to the silencer has been generally given up as unnecessarily complicated and heavy.

The junction pieces of exhaust pipes are generally cast in malleable, or in ordinary cast iron. The latter is cheapest, and seems quite satisfactory.

The exhaust pipes themselves are steel or iron tubes with flanges, which are usually brazed on. If they are close to the cylinder, however, brazing sometimes gives trouble, and then the flange must be screwed on. Copper pipes are not very suitable for exhausts, as the copper gets hot and cracks.

Inlet pipes are generally made castings, very often of aluminium, and can be made a great deal cheaper in this way than if bent up of copper pipe. In many cases it should be almost possible to dispense with piping altogether, particularly when the petrol is fed to the carburettor by pressure, and the carburettor can therefore be put close up to the cylinders. It should, in fact, be possible to cast the necessary branches and connections on the carburettor so that it fits right on to the inlet valve; this would dispense with special brackets. This is done in some cases, but might be done more often.

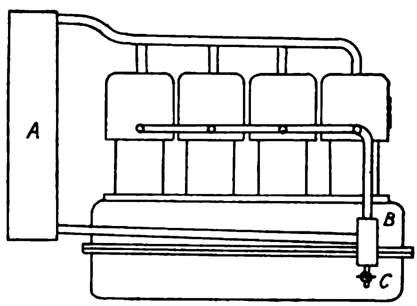


Fig. 187.
A, Radiator.
B, Pump.
C, Drain cock.

The piping for the water circulation is made of copper, with, generally, india-rubber joints to keep it from breaking with the vibration; the connections are made with flanges brazed on, or unions, in the usual way. The piping should be so arranged that all the water in it can be drained out through one cock. This can be done in most cases. In order to make certain of this the inlets for the water into the cylinders must in all cases go well into the bottom of the water jacket. Sometimes it is necessary to have a drain cock on the bottom of each cylinder, which is more expensive, and involves a greater risk of the water jacket being burst by the water left in the cylinder in frosty weather.

If the radiator is entirely above the level of the circulating pump a cock can be put in the bottom of the latter, which will drain the whole of the system, as in fig. 187. If the pump is centrifugal and above the bottom of the radiator this is not so easy, as if the drain is at the lowest part of the piping it does not completely drain the pump, and, if put in the pump, it does not drain all the pipes. It is most important that the pump should be

drained whenever there is a likelihood of the car being in a frost, for, should the water in it freeze, the pump spindle may be twisted off when the engine is started. The remedy is to have two drain cocks. The use of india-rubber piping seems unsatisfactory to an engineer, but it is doubtful if anything better can be used. In some cases the piping has been built up of short lengths with intermediate joints formed by a stuffing box with a gland. This is heavier than the india-rubber pipe and much more expensive, but it lasts longer and looks better.

If there is no pump the pipes must be very large, and there must, of course, be a continuous rise from the top of the cylinders to the radiator, and an entire absence of pockets in which steam or air can lodge so as to stop the circulation. Also, it is necessary that the greater part of the radiator should be above the bottom of the cylinders, as the circulation is due to the difference of temperature between the water in the cylinders and uptakes and that of the water in the corresponding part of the radiator. The pipes should not be less than about 1 inch in diameter for a moderate sized car; they are generally larger.

The petrol pipe, also, is made of copper, and is generally a very small pipe (less than $\frac{1}{4}$-inch bore), but it might be a great improvement to make it larger, as it would be less liable to be choked up, while the extra weight and cost is so small as not to be worth considering. The arrangement of this pipe necessarily depends on that of the petrol tank, but a settling tank for intercepting the dirt in the petrol would be an improvement.

The oil pipes for lubricating are generally small copper pipes, but steel pipes have been used with success, and are probably less liable to crack with the vibration.

With small pipes, especially those made of copper, the unions should be soldered, not brazed, on, as the brazing tends to cause the copper to deteriorate, and to make it much more liable to crack. The union must be made with a very long sleeve on the pipe so that the solder shall have ample surface for making a secure fastening.

Silencers.—In order to make the noise of the exhaust less, it must be taken through some sort of a silencer, so that the separate puffs of the engine may be modified and the gas issue in a more uniform stream. It might be thought, at first sight, that a simple exhaust box with the exhaust pipe going into it and an outlet small enough to make it flow out in a more or less continuous stream would answer all purposes, but, in practice, it does not. In fact, in this case the exhaust box forms a sort of sounding board, which re-enforces the noise made by the exhaust in entering the box. In practice, a great many baffle plates are required in the box for reducing the puffs to a gentle stream.

Every conceivable arrangement of baffles is used, the designer in each case claiming that the silencer gives perfect silence without in any way interfering with the flow of exhaust gas.

Figs. 188 to 191 show some of the more ordinary designs. In practice, the success of these depends greatly on the proportions of the various parts, as any of them can be made to work well if these are right. The usual plan is to gradually reduce the area through the diaphragms successively until the last has an area a great deal less than that of the exhaust pipe.

There is no doubt that cooling the exhaust has great effect in silencing, and that a thorough cooling would be effective. This can be managed if plenty of water is available; but this is impracticable on a car. Still,

something may be gained by cooling the gas as much as possible. Fig. 191, for instance, shows an arrangement by which the gas is kept against the wall of the silencer all the time.

For this reason, also, a silencer of small diameter and considerable length ought to be more efficient than one which is of larger diameter, but shorter. It is easier to fit, and can be made of lighter metal.

The silencer should not be made too light, as, if charged with unburnt gas, the explosion of this will burst it. It should be strong enough to resist a pressure of about 30 lbs. per square inch, which is about the utmost it is likely to be subjected to. Silencers have often been blown to pieces, and should be made to stand something like this pressure, though it is not necessary to give them any great factor of safety.

The dimensions of silencers vary very much; those in bicycle engines are a little larger than the cylinder, while those in cars are often ten times

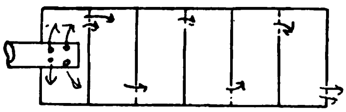


Fig. 188.

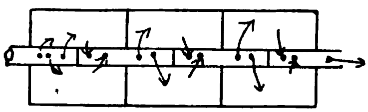


Fig. 189.

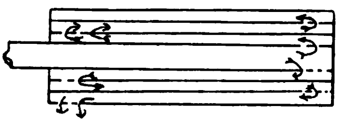


Fig. 190.

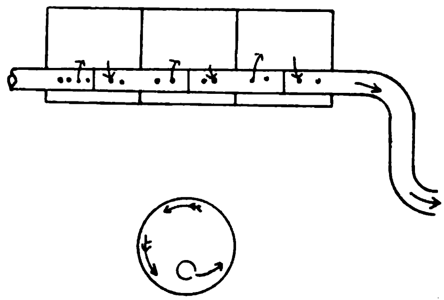


Fig. 191.

the volume or more. It is obvious that the silencer ought to be several times the volume of the cylinder, as the object of it is to make the exhaust issue in an even stream, and, to do this, it must store the exhaust to a certain extent. If it is very small, there must be a good deal of back pressure. Consequently, bicycle engines are generally not silenced at all well.

Silencers are generally made of sheet steel rivetted together. The final exit from the silencer may be either through holes in the side or end, as in figs. 188 and 190, or it may be through a pipe. It should be arranged so as not to blow on the ground, as some of the earlier ones did, as this raises dust. Fig. 191 shows a very good arrangement, the exhaust pipe being carried well below the silencer and then turned backwards. This ensures that the smell shall not run up the back of the body of the car, as it otherwise sometimes does, and be diffused through the back seats.

CHAPTER IX.

GOVERNORS—ENGINE CONTROL—BALANCING.

Governors.—The requirements of a perfect governor are not easy to comply with. It ought to (1) govern the engine as closely as possible to the speed it is set to, (2) govern without "hunting," and (3) be capable of being set to govern over a good range of speed.

The ordinary governor is of the centrifugal type generally used in other engines, and in this case the above requirements are rather antagonistic to each other.

If the governor is set to govern very closely—that is to say, to open full with a very small reduction in speed—it is rather apt to "hunt"; that is to say, first to open full and let the engine run away, and then close entirely, thereby shutting off the gas entirely till the engine slows down a great deal. This makes a very unpleasant running engine. On the other hand, if the governor is set to have a little more range it will not hunt, but will allow the engine to quicken up more when the load is suddenly taken off, as by taking out the clutch.

There is, however, no great difficulty in making a governor to work satisfactorily at a given speed. The trouble is greatest when it is wished to vary the speed. The ordinary way of doing this is to vary the strength of the spring. This works well for a small range, but not for a large one, as it is rather impracticable. The centrifugal force of the governor balls varies as the square of the revolutions, and, therefore, if the governor is to work over a range of from 1 to 10, the spring must range from 1 to 100. Thus, if a spring exerts a pressure of 1 lb. at the lowest speed, it should exert a pressure of 100 lbs. at the highest, which would probably be impracticable.

There are several alternatives to varying the strength of spring, which will be dealt with later.

The ordinary form of governor is shown in fig. 192. In order that this should work well, it is essential that it should be big enough for its work or the friction of the various connections to the throttle will be so great that it will not be able to move it at once. Then it will hunt. Some of the older governors were very small for the work they had to do, but there is a general tendency to improve this. With this form of governor the spring must be one which rapidly increases in tension as the balls fly out; otherwise the governor will not be steady and will hunt. Fig. 193 shows a type which is not so liable to this, as the movement of the balls increases the angle which their supporting arms make with the shaft. This type also allows of much more movement than the last in the same diameter, and, therefore, is generally more convenient to connect.

In either of these types there may be a spring on the governor itself, with outside spring or springs to vary the speed; or there may be one outside spring, the tension of which is varied. The latter seems much the simplest in every way.

There are several ways in which the variation of speed may be made without altering the spring on the governor. The governor may be driven with a variation of the ratio of speed between it and the engine. This would be perfect in theory, as the governor would always run at the same speed, and the engine speed would simply be varied by varying the ratio between it and the engine. It would not be very easy to make a cheap, yet satisfactory, arrangement, but it should be possible.

Another plan is to have a governor which works over a very great range of speed, and to use only a small part of this range at a time. In this case there is a powerful governor with a spring of rapidly increasing tension, so that the whole of the movement of the governor requires the whole range over which it is intended to run the engine. Suppose the engine is to run from 100 to 1,000 revolutions, the governor is so made that it takes this

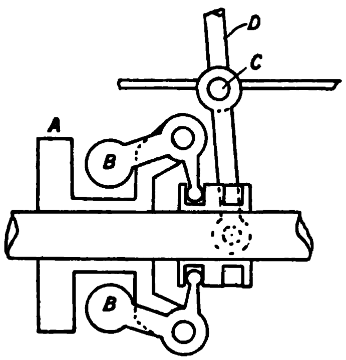


Fig. 192.

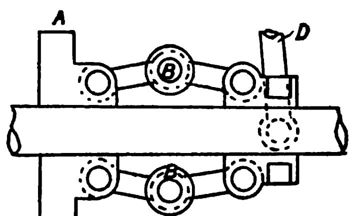


Fig. 193.

A, Pinion of 2 to 1 gear.
B, Governor balls.
C, Pivot on governor casing.
D, Lever to connect to throttle valve.

range of revolutions to make the governor go its whole travel. The connections to the throttle are then so made that a small part of this range will move the throttle from full open to shut. The connections are shown diagrammatically in fig. 194. The movement is the ordinary moving fulcrum, but the same effect could be got with a hunting screw. This, in practice, if well carried out, works very well.

Several other plans of governing have been used, such as the pressure of the circulating water, the pressure of the exhaust, &c. The only one which seems very promising is the pressure of the circulating water.

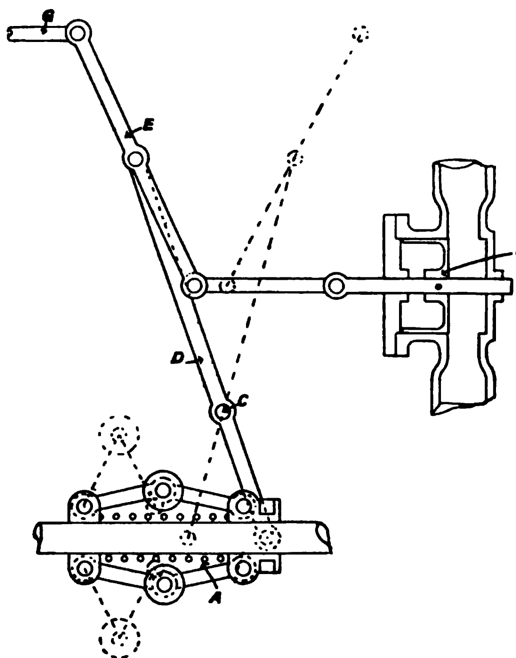
If a positive pump pumps through a contracted orifice, the pressure is proportionate to the square of the speed of the flow, and, if this pressure actuates a diaphragm connected to the throttle, this can be used as a governor. If the size of the orifice is varied, the speed at which it governs will vary in proportion, and, as no alteration of the spring is needed, the same effect should be producible at all speeds.

Engine Control.—There are several different methods of controlling the engine—by an automatic governor or by hand—and each can be carried out in different ways. The following have been used at different times for internal combustion engines of various sorts :—

1. *Cutting out Explosions altogether.*—This was the method of governing

in the early Panhard and many other cars. It was always used in connection with an automatic governor, and with automatic valves. When the speed reached a certain point the governor disconnected the exhaust valve from the runner, so that it did not open. Consequently, the charge was not exhausted, and no fresh charge was sucked in.

Two variations of this are used in stationary work. In one the exhaust, instead of not being opened, is held open. In this case, the inlet valve is automatic with a fairly strong spring, and, therefore, never opens as long as the exhaust is open. This should be better than the last, as the work of compressing the gas at each stroke when the exhaust is kept shut is avoided, as is also the consequent wear on the bearings. In gas engines with a



A, Governor so constructed that its range of movement is 10 to 1—*i.e.*, that it takes ten times the revolutions to make the balls travel to their extreme position, as is required to make them begin to move.
D, Lever pivoted at fixed point, C.
E, Lever pivoted at end of D.
F, Throttle valve.
G, Rod controlling speed at which governor acts.

The throttle is completely opened and closed by a movement of A = one-tenth its total range of movement.

The dotted circles and lines show position at highest speed with throttle shut.

Fig. 194.

separate inlet valve for the gas, the cut-off is, in many cases, only applied to the gas, while the air is allowed to enter as usual. With mechanically-worked inlet valves it should be practicable to disconnect the inlet valve only.

Although the cut-out governor has quite gone out of use for pleasure cars, it seems to have some advantages for commercial work, as it is certainly the most economical, in theory at least.

2. *Throttling the Whole Mixture.*—This is by far the most used at present, and is very simple. A throttle valve is simply put between the carburettor and the engine. The result is that the cylinder is only partially filled at the suction stroke, and the explosion pressures are reduced. The compression pressure is also reduced, and this ought, theoretically, to very much lessen the economy of the engine. Another circumstance that will

affect the economy is that the proportion of burnt gases in the charge is very much greater when the charge is throttled. The lowering of the compression and explosion pressures, however, makes a very quiet-running engine, and, for this reason, it is used for pleasure cars.

3. *Varying the Lift of the Inlet Valves.*—This has precisely the same effect as throttling the charge by a throttle valve. The difference is simply that the charge is throttled close to the cylinder instead of some distance away from it. The length of pipe between the throttle and the engine is so small that it cannot make any appreciable difference in practical working. Some of the early cars had throttle valves close to each inlet valve, but there seems to be no advantage in this, and the plan has not come into general use. A great disadvantage of the variable lift to the valves is that it is difficult to make it exactly the same on all the cylinders. If it is not the same, the engine will not work evenly on all the cylinders when it is running light, and will not run smoothly in consequence.

4. *Varying the Time the Inlet Valve is kept open.*—This has some theoretical advantages over throttling, and there ought to be a slight economy in it. The theoretical economy is, however, very slight, and the complication considerable.

5. *Throttling the Exhaust.*—This is not often done, and can only be with automatic valves. It seems to work surprisingly well, considering the fact that it must entail an enormous proportion of burnt gases in the charge. Probably, in practice, the new charge does not completely mix with the burnt gas, but remains comparatively pure near the inlet valve and ignition plug. This method maintains the full compression at all times, but does not seem to have any sufficient advantages over the throttle to compensate for the extra complication. The throttling of the exhaust is usually effected by varying the lift of the exhaust valves and their time of opening.

Taking it all round, the throttle on the inlet pipe seems quite the best for pleasure cars, though the cut-out may be used in the future for lorries, &c.

In any case, the throttle is the cheapest of all to make and to keep in order.

Governing by altering the time of ignition is extravagant in fuel and is practically obsolete.

Most engines have the throttle controlled by some sort of governor. Much discussion has taken place as to whether this is a necessary fitting or not. It may be said at once that it is *not necessary*, but it is probably sufficiently convenient to be worth the small cost of fitting it.

In the absence of a governor, the best arrangement, probably, is to have a throttle which can be set by a handle on the steering wheel, and a foot accelerator pedal. The throttle is held by a spring, so that when the accelerator is put down it is opened, but comes back to the position it is set to when the pressure on the pedal is taken off. Then, in open country, the engine is worked on the hand throttle, but, in traffic, with the foot accelerator, the throttle being set to the slowest speed at which the engine will run. On taking out the clutch, the foot is naturally taken off the accelerator pedal, and so the engine does not race.

An inverse arrangement to this is often fitted. A throttle is connected to the clutch pedal, so that, at whatever speed the hand throttle is set to, the engine is slowed up on putting out the clutch. This has the disadvantage that the engine cannot be accelerated just before putting the clutch in to start up a hill, as the two are connected. This is avoided if there is a small

pedal for the throttle close to the clutch pedal so that one naturally puts them both down at once, but can use them separately if desired. Both these plans give satisfactory control in the hands of skilled drivers who are used to the car they are driving, but the automatic governor is certainly a great convenience, and is coming into general use.

There are several ways of arranging this. The most perfect is to have a governor which can be set by hand to govern the engine at any speed desired. In connection with this there is generally a foot accelerator to throw the governor out of action when desired. This makes the most perfect control of any, as the hand control can be used in the country and the foot accelerator in traffic. The only difficulty is to make a perfect governor which can be used over a large enough range.

Other means of control are:—

1. To have a governor simply to prevent the engine racing right away when the clutch is thrown out suddenly, and therefore set for a pretty high speed, generally about 1,000 revolutions; and a hand throttle to control the engine at lower speeds. A foot accelerator is generally fitted to throw the governor out of action when extreme speed is of importance.

2. A governor set to run slow, and a hand lever to open it. This is not much used.

A modern engine, while running, should not require any adjustments beyond the control of the speed. The carburettor should be sufficiently automatic not to require any attention while running, and, with a really good ignition, there should be no necessity for altering the timing. With a trembling coil, there is a certain advantage in altering the ignition point as the engine varies in speed, and, in this case perhaps, it is necessary to put the lever controlling it on the wheel with the throttle lever, otherwise there is no necessity. It seems much the best for the maker to fix the ignition point at that which gives the best all-round result, and not to have it liable to alteration.

Whether it should be altered for starting is another matter. With low-tension magneto ignition, there seems no reason for this in moderate-sized engines. Where it is thought necessary to put it later for starting, the best way is to connect it with the starting handle, so that, when the latter is put on, the ignition is put late at the same time.

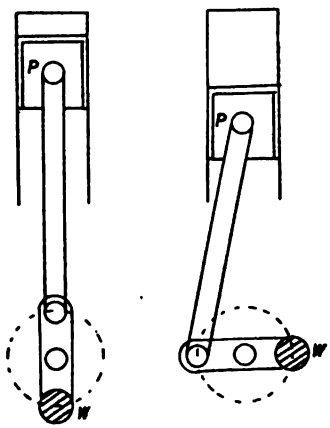
It is now common to have the throttle worked by a lever on the wheel or steering column. The majority of modern cars have a lever on the top of the wheel, with the spindle going through the steering column, and a sector at the top of the wheel. This is probably the most convenient plan possible, and the arrangement will be further discussed in connection with steering wheels.

Balancing.—The simplest case of engine balancing to consider is the single cylinder, which, therefore, will be taken first. There are two balancings here—first, that of the rotating, and, second, that of the reciprocating parts. The first will include the crank, brasses, &c., and the bottom half of the connecting-rod. The second will include the piston, gudgeon pin, &c., with the top half of the connecting-rod.

In considering the question of balance it must be remembered that a weight moving in one direction can only be perfectly balanced by another weight moving in the opposite direction. It will then be understood why it is impossible to perfectly balance any single-cylinder engine, except by having a dummy weight and connecting-rod, which is practically equivalent

to having a two-cylindered engine, or some other special arrangement. The crank and rotating parts are easily balanced, as all that is necessary is to have suitable balance weights opposite to them. On the other hand, the piston and reciprocating parts cannot be perfectly balanced by any rotating weight, as they are not always moving (fig. 195).

In order to perfectly balance the reciprocating parts at half-stroke there must be an equal weight on the opposite side to the crank. The position here is, in fact, exactly as if the reciprocating parts were rotating, as they are moving at the same speed as the crank. Now, when the position at the end of the stroke is considered, it is seen that the balance weight itself is not balanced. Taking a vertical engine, it is found that if the reciprocating parts are balanced for the vertical vibration, there will be an equal amount of lateral vibration. The best all-round balance is obtained when the balance weight is equal to that of the rotating parts and of *half* the reciprocating parts. This gives half the vibration in either direction. If the vertical vibration is found to be more objectionable than the horizontal the



If P is a reciprocating weight, and W a weight to balance it, W = P.

W is quite unbalanced when P is at the top of the stroke, and the engine is no better balanced in this position than it would be at half-stroke with no balance weight to P at all.

If $W = \frac{1}{2} P$ then there is an unbalanced moment $= \frac{1}{2} P$ at any part of the stroke.

If R be rotating weights and P reciprocating parts, the best balance is when $W = R + \frac{1}{2} P$.

Fig. 195.

balance weight should be made heavier than this, and lighter if it is less objectionable; but vibration in both directions cannot be eliminated.

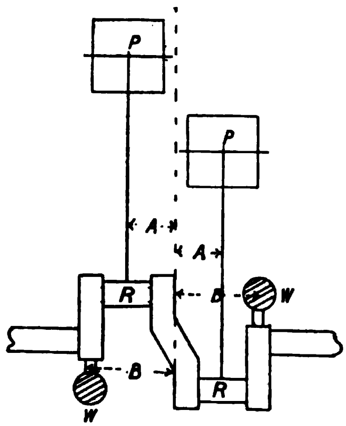
If it is not convenient to make the centre of gravity of the balance weight the same distance from the centre of the crank as the centre of the crank pin, the weight must be increased or reduced inversely as the distance varies. In other words, if W is the weight of the balance weight; R, radius of centre of gravity; w, weight of crank pin; r, radius; then $(W \times R) = (w \times r)$.

In order to get the balance perfect endways the centre of gravity of the balance weight must be in the same line as the piston; that is to say, in practice there must be two balance weights of equal weight, one on each crank. Single-cylinder engines with outside flywheels have been made in which the balance weight was on the flywheel, but this will introduce a bad rocking moment. If there are two cylinders in line the above rules are equally applicable.

If the two cylinders are in line and the cranks opposite, the engine, as a whole, is balanced; since one piston counterbalances the other. There is

a slight rocking moment, due to the distance apart of their centres. If each cylinder is balanced as a single cylinder, this disturbance will be eliminated as far as possible. It is not possible to eliminate it absolutely, since, when the vertical rocking moment is got rid of, a horizontal one is introduced, due to the balance weight itself being unbalanced at the top and bottom of the stroke.

If there is no bearing between the cranks it is only necessary to use one balance weight on the web furthest from the centre of the engine. The



Balance weight to balance rocking moment of two-cylinder engine with cranks opposite.

W is balance weight.

P, Reciprocating parts.

R, Rotating parts.

A, Distance of centre of cylinder from centre of engine.

B, Distance of balance weight from centre of engine.

$$W = (R + \tfrac{1}{2} P) \times \frac{A}{B}.$$

Fig. 196.

weight of this may then be deduced from that necessary to get the best balance for a single-cylinder engine in proportion to the distance from the centre of the engine divided by distance of centre of cylinders from centre line of engine (fig. 196).

A two-cylinder engine with the cylinders opposite and the cranks opposite would be perfectly balanced, but for the very slight rocking moment due to the centres of the cylinders not being absolutely opposite longitudinally. This is a negligible amount, and the engine is practically

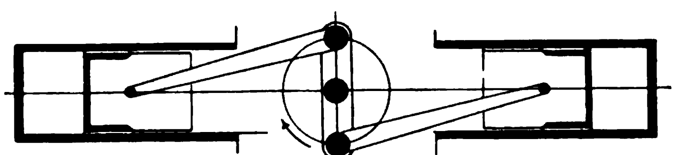


Fig. 197.

better balanced than almost any other engine in use. If four cylinders are used, there being two opposing pairs, the balance can be made quite perfect, as the rocking moment can be made to work in opposite directions, as demonstrated by the extensive employment of horizontal engines of this type in the United States and elsewhere (fig. 197).

Another two-cylinder engine which is very well balanced is the two-cylinder diagonal (fig. 198). In this the cylinders are at right angles, and, therefore, the one piston is standing still while the other piston is moving at

its fastest, and as the one accelerates the other decelerates. To make this engine perfectly balanced the whole of the rotating parts must be balanced and the reciprocating parts of one cylinder. Then the balance weight will balance the moving parts, for all practical purposes, perfectly in all positions.

The three-cylinder engine, like the two-cylinder with cranks opposite, is

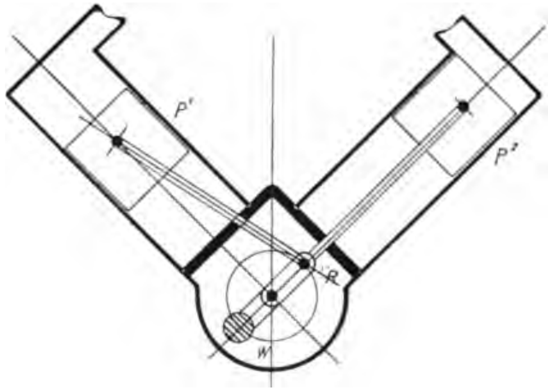


Fig. 198.

balanced as a whole, but has a rocking moment; and, as the centres of the end cylinders are further apart than those of a two-cylinder engine, this moment is larger than in the case of the latter. If each cylinder is carefully balanced it is not a very important amount in any ordinary engine used in a car.

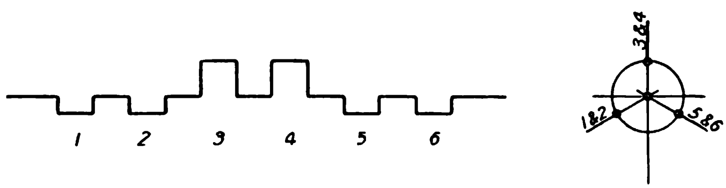


Fig. 199.

The four-cylinder engine with the two middle cranks together and the two end cranks together is perfectly balanced.

The six-cylinder engine has been made with several different arrangements of shaft.

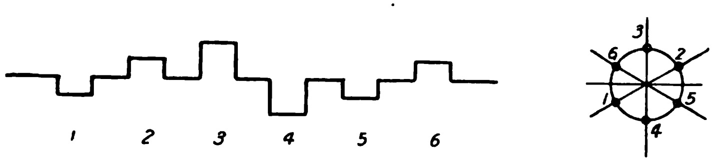


Fig. 200.

1. *Each adjacent pair of cylinders may be together* (fig. 199).—This is cheap and convenient to make if there is no bearing between adjacent pairs of cranks. It gives the same type of balance as the three-cylinder engine,

but as the engine is much longer the rocking moment is much more serious.

2. *Each adjacent pair may be opposite* (fig. 200).—This is very nearly balanced, the rocking moment of the adjacent cranks not being very important, as they are very near together and the engine as a whole is of considerable length.

3. *The crank may be arranged as in fig.* 201, in which the two middle

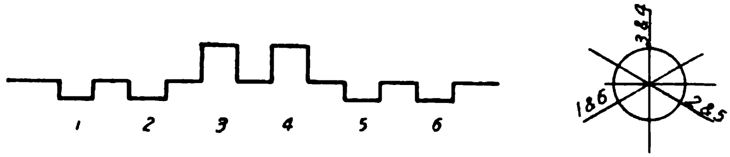


Fig. 201.

cranks are together, the two end ones together, and the two intermediate ones together. This is equivalent to having two three-cylinder engines in one with their rocking moments in opposite directions, and consequently is perfectly balanced.

Although in some cases the engines are spoken of as being perfectly balanced, this would really only be so if they had connecting-rods of infinite length.

If an engine is balanced by the pistons moving in opposite directions it will only be perfectly balanced if they move at equal speed. With an infinitely long connecting-rod this would be so, but, in practice, the arc of the connecting-rod introduces an error, as the piston moves much faster during the top half of its stroke than the bottom. This is indicated in fig. 202, which shows the effect of a short connecting-rod; as will be seen,

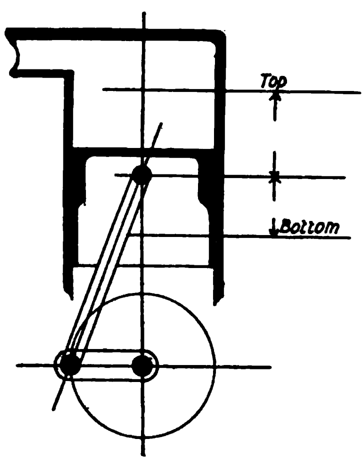


Fig. 202.

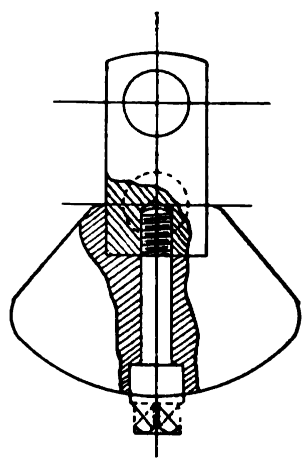


Fig. 203.

the movement of the piston during the top half revolution of the crank is nearly half as much again as during the bottom half. This error is reduced by making the connecting-rod longer, but in engines with all the cylinders

in line it cannot be completely eliminated, as the balancing of the piston at the top of the stroke is always effected by another piston at the bottom.

In an engine with the cylinders opposed, however, the error is balanced, as the balancing of one cylinder with the piston at its furthest from the crank shaft is effected by another, also with its piston at its furthest, but moving in the opposite direction.

When balance weights are used the best way of fixing them on is to make them in one piece with the crank shaft, except in the case of a built-up crank. This is not much more expensive in small engines than fixing them on, especially if there is a large flange to the crank shaft to which the flywheel can be fastened. Fig. 203 shows a method of fixing which gives good results if really well fitted, but the bolt must be a good fit both in the plain part and in the screw, and the jaws must fit well on the crank.

It will be seen that in the two-cylinder engines the only type which gives an even division of explosions and balances the moving parts is the opposed type; generally speaking, this is not very convenient to arrange and is probably rather expensive to make.

Of the other types the different turning moments are given in Chapter iii. In considering the relative merits of two-cylinder engines of the types with the cranks together and that with the cranks opposed, it is a question whether it is better for the explosions to be at regular intervals or the moving parts balanced. This depends largely on the weight of the moving parts and the speed at which the engine is run. The faster it runs the more continuous are the impulses, but the vibration from unbalanced parts is greater. On the other hand, with slow speeds it is important to divide the impulses evenly, while want of balance is not so apparent.

Besides this, the lighter the moving parts are kept, the less the vibration from them is. Therefore, it appears at the present time, when all engines are made with very light moving parts and are generally run at moderate revolutions, putting the cranks together is much the best plan.

The custom of putting the cranks opposite was introduced when pistons were made far heavier than they are now and was no doubt necessary with these to prevent excessive vibration. With light pistons, &c., and good balancing, the engine with the cranks together seems in every way the best at any speed up to which modern motors are habitually run.

CHAPTER X.

SPECIAL MOTOR ENGINES.

Motor Bicycle Engines.—These have a general resemblance to other engines, the most important points of difference being the relatively greater importance of cheapness and lightness, and that the engine is air-cooled. The future success of the motor bicycle will probably depend on its being kept light and a distinct type of machine from a car. If it becomes like a two-wheeled car it will cease to have the merits either of the car or of the bicycle.

On the other hand, as long as the machine is light, absolute reliability is not so important as in a car, for the simple reason that it can always be pedalled to a railway station, whereas with a broken-down car further progress is checked. As lightness and reliability are opposite requirements, it may, therefore, be wise to run slightly more risk of breakdown than in a car.

Lightness of weight generally involves increasing the number of revolutions in a given time as much as possible, and, therefore, larger valves are needed than in cars.

Air-cooling.—The air-cooling requires the cylinder design to be considered from a point of view rather different to that of a car engine. Considering that the power of an engine is dependent on the differential range of temperature of the charge before and after ignition, its initial-temperature should be kept as low as possible, consistent with its being able to explode. Means must, therefore, be taken to minimise the heat it receives from the cylinder walls as much as possible by a suitable arrangement of the valves.

Valve Arrangement.—This is to a great extent governed by the mechanical requirements of the engine, as it is necessary that the design should be a workable one.

Figs. 204, 205, 206, and 207 represent different arrangements of valves in use, and the direction of the inflowing currents of gas. In figs. 204 and 205 the contact of the gas with the hot surface is considerable, but less in fig. 206, and still less in 207. Another consideration is that of the proportion of surface of cylinder to compression volume, noticed in Chapter vi. which is of importance for the same reason—that, if great, it heats the charge. Still it seems likely that this is not so important as the heating of the charge on coming into the cylinder, as it does not take place to any great extent till near the ignition point. The heating of the charge before compression, on the other hand, reduces the weight taken into the cylinder and, consequently, the explosion pressure.

The arrangement with both valves at the top gives the best shaped combustion space, and also a very good arrangement of inlet. On the other hand, the arrangement with the exhaust at the side and inlet over the cylinder gives a better inlet still, and allows of the exhaust valve being

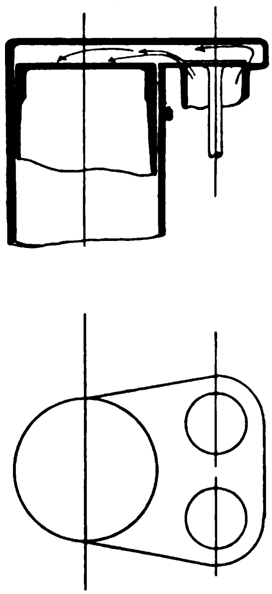


Fig. 204.

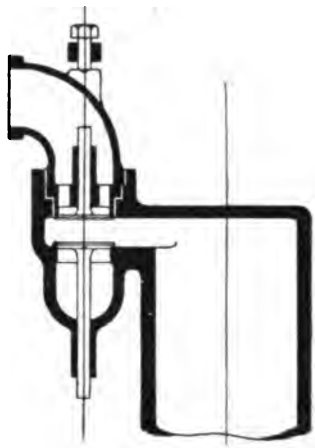


Fig. 205.

Fig. 206.

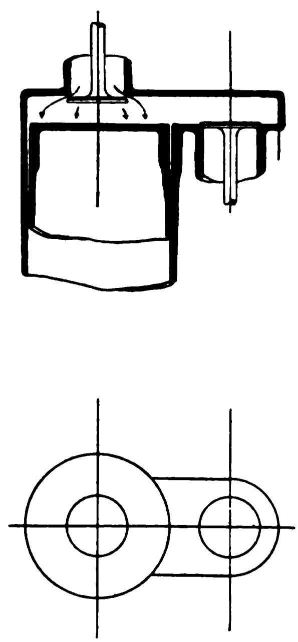


Fig. 207.

direct driven. With both valves at the top it is difficult to make them large enough. The best arrangement is with both valves at the top at an angle, but is not easy to manage.

The question of automatic as against mechanically-worked inlets is another point in which the bicycle may be different from the car. Although there has been a great run on mechanical valves lately for bicycles, there seems to be no advantage in them, and, in some cases, makers have returned to the automatic valve. The latter is distinctly cheaper and slightly lighter, while it has fewer moving parts. As it is not convenient to put the mechanical valve on the top of the cylinder, it is generally placed at the side which makes a badly shaped cylinder and inferior valve arrangement. In fact, the mechanical valve was often adopted without due consideration of the differences between cars and bicycles.

In practice several engines, in which the inlets have been at the top

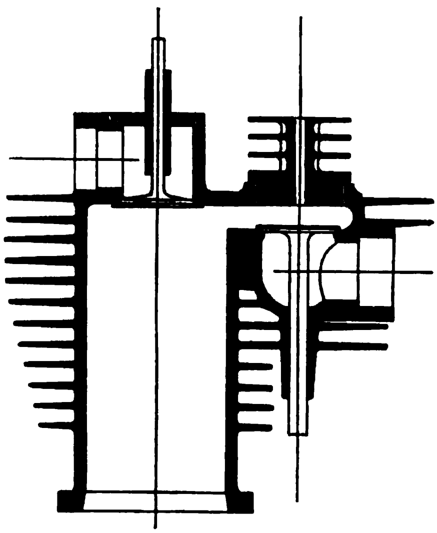


Fig. 208.

of the cylinder, have been remarkably successful in developing the maximum power of a small and light engine.

The air-cooled cylinder must have ribs cast on it to radiate the heat, but otherwise much resembles those that are water-cooled. The ribs should be parallel with the current of air. There are doubts as to what depth is best. At first sight it would seem that the deeper they are the more they would cool the cylinder, but in practice there is a distinct limit. They also add to the weight.

Fig. 208 shows a cylinder I have used which has the valves at the side and top, and with very deep ribs. These ribs in actual running were often cool enough at the edges to be touched with the hand, so that the outside portion cannot have had much influence. Probably the heat is dissipated by radiation from the cylinder more than from the ribs. Generally ribs extending about $\frac{3}{4}$ inch from the body of the cylinder are quite deep enough.

The valve shown in fig. 205 is inserted from the inside of the cylinder,

so that when replaced by a new one the cylinder must be taken off. The absence of a joint is an advantage, as small engines lose power from leaks far more than large ones, but the arrangement is very inconvenient, and a detachable valve seat is better; one of the latest racing cars, however, has a valve put in the way above described. The great advantage of putting the valves over each other is that there is only one joint for the two valves.

The cylinder is now generally cast in one piece, the practice of casting the head separate having almost ceased. The former is in every way the best and has the further advantage of avoiding a chance of leakage.

Flywheel.—The majority of bicycle engines have the flywheel enclosed, as in the case of single-cylinder car engines. It is the most convenient way, but, with an outside flywheel, the engine, the crank case, and the flywheel can all be made lighter. The flywheel can have a larger diameter,

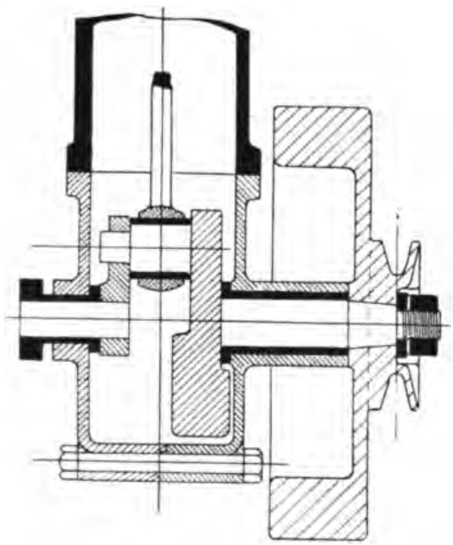


Fig. 209.

thereby also increasing the effective power of the engine and decreasing the liability of the belt to slip.

A common plan of making the crank case when the flywheel is outside is to have a forged crank running in one long bearing, as shown in fig. 209, there being a dummy crank on the other side to drive the 2 to 1 gear. The difficulty in this, as in most arrangements with outside flywheels, is to make the engine narrower than the space between the pedals.

As the engine is to run very fast it is more important to study the weight of the moving parts than in a car; this is best done by having two rings only in the piston, as is often done.

The stroke of bicycle engines is generally about equal to the diameter, as it is desirable to keep the engine short and light.

Ignition.—The ignition on bicycle engines has usually been high tension either with or without a magneto. The accumulator and coil is the cheapest form, and this has no doubt had a great deal to do with its popularity.

Why the low tension has not come into use is not quite clear. It was used in the Singer machines with the engine entirely inside the wheel, as there it was absolutely necessary that the ignition should be perfectly reliable, and it seems that it actually was so. It seems very doubtful whether any form of high-tension ignition would have worked at all in a place so liable to be covered with mud and water.

In many other cases the low tension was badly planned and the ignition plug made so small that it was impossible to obtain proper insulation. This no doubt tended to give the principle an undeservedly bad name. The fact that a high-tension ignition can be added to any engine without any special care being required in designing it is perhaps also a factor. A good low-tension system of ignition should be a great improvement, however, as the high tension is always liable to be a source of trouble.

If high tension is adopted the ignition must be by single spark, not by trembling coil, as there is a marked loss of power with the latter at the high speeds these engines run.

Number of Cylinders.—Most bicycle engines of moderate power are made with one cylinder as this is the simplest and cheapest. There are many advantages in making the higher powered ones with more. In the first place, the weight of the flywheel is much less. In fact, the flywheel necessary for a two-cylindered engine is less than that for a one-cylinder with the same sized cylinder and, consequently, only half the power. Then for a given power the impulses are much smaller, and so the tendency of the belt to slip is less. Also the tendency of the tyre to slip on the ground is less. This latter is a matter of importance with high powers as there is a considerable increase in the wear on the tyres.

There are several ways in which the two-cylinder engine can be arranged. The cheapest is to put them diagonally on one crank with inside flywheels, in which case the weights are well balanced; but the engine is awkward to get in, while one of the cylinders shields the other from the wind.

In some cases the cylinders have been put side by side, as in fig. 210; this divides the impulses evenly and makes a neat arrangement. The engine with two opposed horizontal cylinders has also been used and makes a very neat arrangement, which has many advantages, but it necessitates a forged crank and split brasses to the connecting-rods, and is therefore, perhaps, too expensive.

A type of cylinder with some advantages, which was used in racing tricycles years ago, is shown in fig. 211. In this there is a row of holes through the cylinder wall level with the bottom of the piston stroke. These are uncovered just at the end of the working stroke and also of the suction stroke. The advantage claimed for it was that it allowed of a freer exit for the exhaust gases. This may have been so, as the engines of those days often had valves much too small for the speed they ran, but it is obvious that the same result can be obtained by making the exhaust valve of suitable size. There is, however, another advantage in this construction—viz., that the cylinder is always completely filled at the end of the suction stroke. The result will be a stratified charge in which the rich mixture will be round the ignition plug and pure air above the piston. Thus we have a mixture with a low proportion of gas, while that round the ignition plug is rich enough to ignite. This should make for an economical engine which, as experience proved, ran with very little heating. The objection to this engine was its noisiness.

Possibly the same result might be obtained in another way. There are stationary engines at work in which the cycle of operations is similar to that indicated in fig. 212. In this a charge of air is sucked into the crank case through an automatic valve (not shown) on the upward stroke of the piston. This is slightly compressed on the down stroke, and when the piston overruns the ports connecting the two it rises into the cylinder. This, in the engine as shown, would happen at the end of (1) the working stroke, and (2) the suction stroke.

At the end of the working stroke the exhaust valve would open just before the ports were uncovered and the pressure would fall to that of the atmosphere. While the crank was rounding the bottom centre the air which

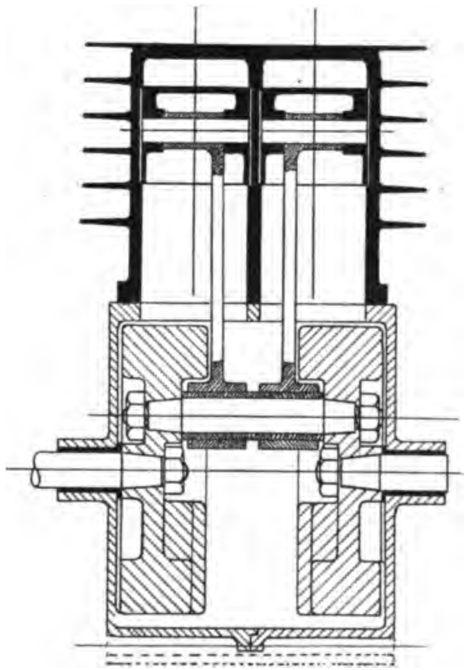


Fig. 210.

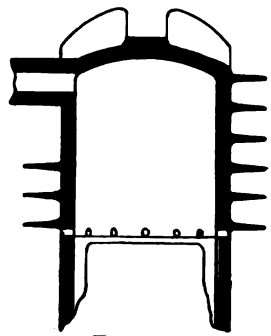


Fig. 211.

had been previously drawn into the crank case and slightly compressed would fill the cylinder and drive the exhaust gases out of the exhaust pipe so that the hot gases would be got rid of, and the cylinder at the end of exhaust would only have a residue of pure air. It is well known that, if we get rid of the exhaust gases from any explosive mixture, the power of the explosion is increased and the cool air flowing through the cylinder during the exhaust stroke helps to cool it.

Again, the cylinder gets a larger charge, for at the end of the suction stroke air, under slight pressure, would flow into the cylinder, so that the compression stroke would start with a charge a little above atmospheric pressure instead of one a little below. As the power depends mainly on the weight of the charge in the cylinder this is a perceptible gain. This, as far as I am aware, has not yet been tried, but should be quite worth trying.

In the magneto ignition it seems a pity that we do not always fix the magnet in the flywheel. This would at once reduce the weight of the machine by the weight of the magneto, as the flywheel with magnets in it would be no heavier than one without.

Two-cycle Motors.—The only motors considered hitherto work on the ordinary Otto four-stroke cycle. There are, however, motors working on the "two-stroke" cycle, in which there is an explosion every revolution.

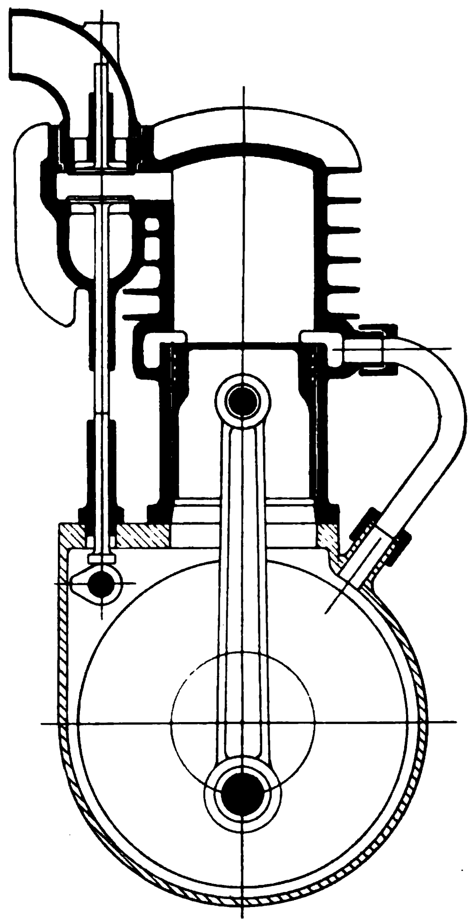


Fig. 212.

This arrangement is a very old one, having been used in the Clerk gas engine thirty years ago, but, in this form, it has not come into use for such work as petrol motors. For these it has generally been modified, as in fig. 213, which shows the general principle.

The action is as follows:—On explosion taking place, the piston is driven down, and the charge of mixture in the crank case is slightly compressed.

When the piston reaches near the bottom of its stroke, it uncovers an exhaust port which lets out the greater part of the exhaust gases. As it continues to descend it uncovers a port connecting the cylinder with the crank case. The gas which has been compressed in the latter flows through

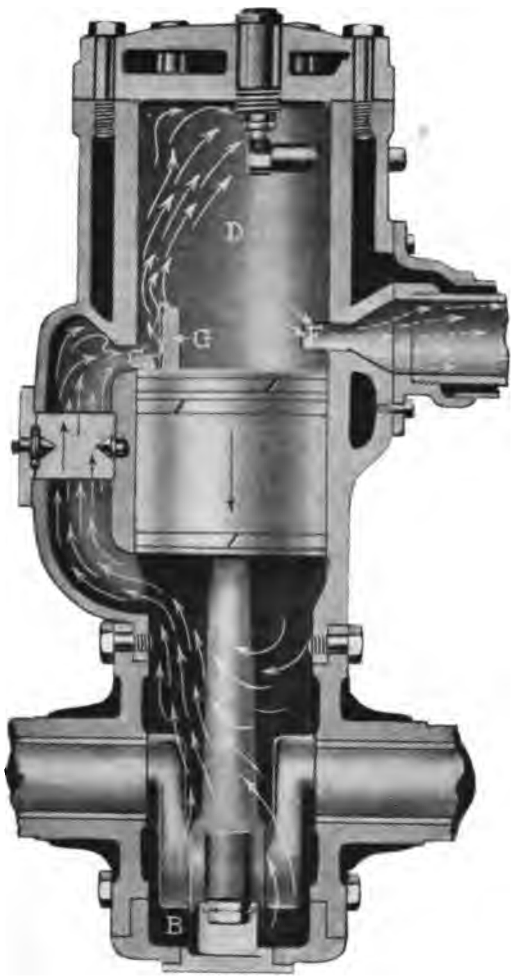


Fig. 213.

this port, is deflected into the upper part of the cylinder, and drives the remaining exhaust out of the exhaust port. The piston then returns, sucking a fresh charge of gas into the crank case through a non-return valve (not shown), and compressing that in the cylinder, which is fired at the top of the stroke in the usual way.

This type of engine has been used very extensively in the United States

for small marine work, and has many advantages. In the first place, it has a working stroke every revolution. This means that, *with a given number of revolutions and cylinder pressure*, it gives twice the power of the four-stroke. The difficulty is that one can *not* in practice get the same pressure or run at the same revolutions, as explained in next paragraph. It also has the advantage that the pressure is always on one side of the bearings, and consequently slack, and this does not cause it to knock. It is very simple, and the only valve is the one in the crank case, and this is sometimes dispensed with.

With these advantages it is rather surprising at first sight that it has not come more into use. It is often stated that, for a given-sized cylinder, it would yield twice the power of the four-stroke, as it has a working stroke every revolution instead of every other revolution. This, as a matter of fact, is an error, for the power yielded by the two-stroke cylinder is generally far less than that from a good four-stroke cylinder of equal size, owing to the impracticability of getting so high a mean pressure and so large a number of revolutions. The short time available for exhausting and refilling the cylinder at the bottom of the piston stroke prevents this operation being perfectly performed at high speeds, and, consequently, the explosion is feeble. Another difficulty is that the gas may be still burning when the port connecting the crank case with the cylinder opens. In this case all the gas in the crank case explodes, and the engine has to run several revolutions before there is another full charge for an explosion. For these reasons both the revolutions and cylinder pressures must be much lower than the four-stroke, and the cylinders be large for their power. Under these circumstances they run very smooth, and, for very small marine motors, have a great sale. For large stationary engines, this type is also coming into use again, as large cylinders in proportion to their power are not a great disadvantage, while the absence of valves, to be clogged up with tar, is a great advantage.

Some of the disadvantages of this type for motor work may be overcome in the future, and it is certainly a fascinating type to study. The explosions in the crank case can be avoided by mixing the petrol and air on the way from the crank case to the cylinder instead of before entry to the crank case. The difficulty of getting rid of the exhaust might be removed by blowing an excess of air through, as is done in stationary engines, but this generally entails a separate pump.

A point to consider, also, is that it is very difficult to completely fill the cylinder without losing some of the fresh gas out of the exhaust port. In fact, this depends entirely on the incoming charge not mixing with the exhaust gases, and this cannot be positively certain. If the fresh charge of air was sufficient to completely wash out the exhaust gases, and the fuel was then injected into the cylinder afterwards, there would be no difficulty.

Marine Motors.—There is evidently an enormous field in the future for the internal combustion engine in marine work, as indicated by its extensive use for small work by engines using petrol (called gasoline in the United States). The demand for a simple and reliable motor is simply astonishing.

In Europe the development has largely followed on car lines; and racing has been allowed to dominate the industry to such an extent that while some firms have based their reputation on the reliability of their machines, their efforts have been somewhat impeded by the large demand for flashy

racing machines and for boats which have made record speeds, but lacked durability. In the United States, on the other hand, racing has played little part in the development of the marine motor; consequently, it is in use for all sorts of purposes in very large numbers, and is made by the thousand. Hence, owing to their greater reliability, the trade with our own Colonies is almost monopolised by the American makers.

There is an easy remedy for this state of things, viz., to make good motors for marine work and not to fit a boat with car motors. This last practice has done an enormous amount of harm to the marine motor industry in this country, for the requirements are very different.

In a car the engine is seldom run at its full power. It should be very light for its power, and be capable of running over a great range of speed, as also of working with a great variation of torque; as a rule, the greatest number of revolutions will be with the least torque and *vice versâ*. The main point, however, is that it is not required to work at either its highest number of revolutions or greatest torque for long at a stretch. In a boat all these conditions are quite different. The ordinary motor will have to run at its full torque and revolutions all the time. This means that all the bearings, &c., will get far harder work in the boat than on the car.

The difference between the two is far greater than many people imagine. There are very few car engines which, in daily running, do more than half their rated power, taken at the average of a run of, say, two or three hours. This can be very easily seen from the petrol consumption. To take a specific instance—viz., the Tourist Trophy Race, 1905, in which no car burnt more than about 10 pints an hour. More than about 1 horse-power for each pint per hour burnt could hardly be expected, so that the highest power developed for the whole run probably averaged only 10 horse-power, although occasionally it might have been nearer 30. The cars were certainly driven a great deal harder than is usual in ordinary touring, as the course was open and without hindrance. It may safely be assumed that the ordinary car is seldom called upon to exert continuously more than a fourth of its maximum power.

This is a very important point to bear in mind in all considerations with regard to cars when we compare the size of the various parts, &c. It is often said that these are wonderfully light for the work they are expected to do, but in many cases the work they actually do is very much over-estimated.

In a boat, on the other hand, engines may be run at full speed for hours in the case of small boats, and for weeks in the case of large ones; in any case they have to run for many more hours in a year than a car engine; consequently, the bearings, seatings, &c., must be much more substantial than those of the car engine. For this, among other reasons, cast iron will be a better material for the crank chamber than aluminium.

For these and other reasons it will rarely be desirable to run marine engines at as high revolutions as is usual with car engines. As mentioned, the car engines are only run at their full number of revolutions for very short times at a stretch, and then only with a very small load, whereas the marine engine always has the greatest load with the highest number of revolutions. Another point is that noise and vibration are far more apparent in a boat than in a car. As these increase with the revolutions the number of them must be more limited. For marine steam engines, which run at all continuously, anything above 600 feet of piston speed in the

smaller sizes is exceptional, and for years this has not been exceeded, except in special cases. For motors the same limits will probably apply as to any other form of marine engine, and it will not be wise to go much beyond this.

Lubrication also requires more attention in the marine than in the car engine for the same reason as the bearings are larger. It is very doubtful if the plan adopted in the car of simply pouring oil into the crank case will answer for a marine motor. Probably, for the larger sizes at all events, forced circulation will be the best plan. There will be much less objection to the motor emitting smoke from the exhaust, as this can be turned up a funnel.

As stated above, the best material for crank cases is cast iron, not aluminium. The former is far stiffer and, therefore, makes a much more rigid seating for the bearings. It is for this reason that it is almost always used for marine steam engine bedplates and frames. Further, aluminium is liable to be attacked by salt water.

In general construction, small marine motors will be much the same as the car motors, but the whole of the bearings, &c., should be carried on the lower half of the crank case. The arms for securing the motor in the boat should also be in this half. The crank case should also have doors large enough and in the right place for the main bearings, &c., to be got at and adjusted from above without having to take the motor out of the boat. Otherwise the car motor and the smaller boat motor will differ but little.

It is, however, quite clear that there is going to be a very large field for the marine motor in sizes which are far larger than any motors which are likely to be put into cars. Petrol motors have been built with six cylinders 14 by 14 inches, developing over 400 horse-power. In fact, the *Gregory*, which came across the Atlantic on her own bottom, had twelve cylinders, 12 by 14 inches, and developed about 700 horse-power. The engines were on twin screws, each engine having six cylinders, and were reversible, being reversed and started by compressed air. Petrol, however, seems hardly likely to be a suitable fuel for the larger sizes, and producer gas seems to have greater possibilities. In this respect less has been done at sea than with stationary engines. Most of these are horizontal engines of quite different design to the marine motor, but there have been several engines built which are vertical and generally of the marine type.

Messrs. Campbell have built four-cylinder engines with cylinders 19 inches in diameter by 21 inches stroke which develop 360 horse-power, and which have been in continuous use for some considerable time with satisfactory results. The Westinghouse Co. have built engines with three cylinders, 34 by 60 inches, developing 1,500 horse-power. In fact, there seems no reason why very large marine engines should not be built. Messrs. Beardmore & Co. have under construction a marine gas engine with five cylinders, 20 by 24 inches, which is expected to develop over 500 horse-power in continuous working, but at the time of writing this has not been tried. In the immediate future, however, there will not be much done in sizes as large as this. Still, in engines with cylinders exceeding about 5 or 6 inches there will be many points which will require careful attention in the design if it is to be satisfactory. In the first place, the design should be such that all repairs and renewals can be effected without taking the engine out of the boat. In many cases engines of this size will be placed below deck, and it would not be at all

convenient to take these out for small repairs. They should, therefore, be so arranged as to allow of every bearing being adjusted or replaced with the engine *in situ*. Even the crank shaft should be removable, which could be made possible by having very large inspection doors, supporting the engine on columns, and making the doors as long as the crank case.

It is very desirable that all working parts should be accessible from one side of the engine, and that the cam shafts, &c., be withdrawable from the same side. If they are threaded out endways the engine-room must be twice as long as the engine.

The cylinder design will be, to a certain entent, modified by these considerations, for it is obvious that an engine that has its valves on opposite sides would not suit this condition. Besides this, it will be observed in examining the cylinders in figs. 101, 102, and 103 that the caps above the holes occupied by the valves are not water-jacketed. The caps could be cast hollow and be jacketed, but this is not at all convenient, and, even then, a very complicated arrangement of pipes, difficult to drain, would be needed for circulating water through them. Therefore, some design of cylinder should be chosen which avoids this. In addition to this it is necessary, if producer gas is used, that the cylinder should be easily accessible for cleaning tar from the top of the piston, &c.

Details bearing on the modification of compression, water-cooled valves, and other things necessary in the larger motors using producer gas will be found in works on gas engines.

It is not decided if a marine motor requires a crank case. The large majority of the smaller ones are cased in, and with good reason, but it is a question if the larger engines should be. The car engine must, of course, be cased in to preserve it from the dust, but there is no dust in an engine-room at sea. Most marine steam engines are still built open, a plan which seems to have many advantages. In the first place, the engine is lighter, as the casing is saved. Then, being more accessible, it can be examined when running, and the bearings felt; if these are too warm water can be poured on them. Any repairs or adjustments that may be required are much more evident and easier to carry out. On the other hand, if the engine is cased forced lubrication can be applied to the bearings.

It is noteworthy that the *Gregory*, which is the only high-speed motor boat to cross the Atlantic on her own bottom, has open engines carried on steel columns. Still the general tendency of engine building is to case the engines in, though this practice is extending very slowly.

A very important point in the design of marine motors, except in the very smallest sizes, is the matter of starting and reversing. In the small sizes this is done by having either a reversing propeller or a reversing gear worked with clutches, the engine being always kept running, and generally controlled by a governor. Both the reversing propeller and reversing gear, however, present certain difficulties when made of any size and used for continuous hard work at sea, so that it may then be necessary to have an engine which can be started and reversed like a steam engine. There is no impossibility in this, and such engines have actually been made in the United States to a considerable extent.

For this purpose the action of the cams must be reversed, so that the timing of the valves shall be right for the engine running the reverse way. There are three general ways of doing this.

1. By having two sets of cams, either of which can be brought into action as required.

2. By reversing the motion of the cam shaft relatively to the engine.

3. By reversing the direction of flow of gas in the inlet and exhaust pipes.

The first is, generally speaking, considerably more complicated than the last two.

In order to start the engine there must be a reservoir of compressed air and valves for letting this into the cylinders at the right time.

There are several suitable arrangements, but to make sure that the engine can be started the cylinders must be so arranged that there is always one piston to which the compressed air is admitted on the working stroke. With a four-cylinder engine as usually arranged, this is not the case; but it is so with either a four-cylinder engine with one pair of cranks at right angles to the other pair, a three-cylinder, or a six-cylinder engine.

With the first two of these it is necessary to have some arrangement for altering the setting of the exhaust valves in order to start, as these will have to exhaust every stroke while starting instead of every other stroke. With the six-cylinder, on the other hand, no such arrangement is necessary, as there always is a cylinder working. All that is necessary for starting a six-cylinder engine is to have a valve arranged for letting compressed air into the cylinder when in working position. This air must be compressed to a pressure well above the compression of the engine, and it will then run the engine long enough for it to start running on its own explosions. For engines of any size the compressed air will be furnished by an air compressor worked by a small motor.

In general, it is absolutely necessary that a marine motor should be reliable. On a road, if the motor stops, the defect can generally be repaired without the stoppage being of serious consequence; but at sea the stoppage of the engine often means the loss of both the boat and the crew. Under these circumstances reliability is the first consideration, and speed of much less importance. The ignition, therefore, must never fail, nor be liable to short circuit through damp. There should be no attempt to make a marine motor as light as a car motor.

CHAPTER XI.

PARAFFIN CARBURETTORS AND GAS PRODUCERS.

Paraffin Carburettors.—Paraffin has often been recommended as a substitute for petrol in the motor, mainly on the ground that petrol is likely to become too expensive, and that paraffin is cheaper. The probabilities are that the supply of petrol will be fully equal to the demand, and that as this increases its price will be lower than at present. Paraffin is cheaper and less dangerous than petrol, but the latter [is by far the cleaner to handle. Paraffin may be preferred for commercial motors, on account of its cheapness, and petrol for pleasure cars, on account of its cleanliness.

If, however, any class of vehicle burning paraffin is to be really satisfactory, it must be so arranged that it runs under all ordinary conditions without emitting any smoke or unburnt oil. In marine work this is not so absolutely necessary, as the exhaust can always be turned up a funnel like the smoke of a boiler. This could be done in lorries, &c., and would certainly be better than discharging an objectionable exhaust on the ground, but even then much smell would be objectionable. Stationary engines burning ordinary paraffin have been in use for many years, but the exhaust has seldom been good enough for motors.

Books on gas and oil engines should be consulted for what has been done in stationary engines. The older engines had some form of spraying arrangement to spray the oil along with all or part of the full proportion of the air, and a vaporiser to heat this. The mixture thus formed was drawn into the cylinder and fired after compression. This plan is still used to a certain extent, but it is now more usual to draw the oil and air into the cylinder, and vaporise the oil inside the cylinder by having the portion against which it strikes unjacketed.

It is not possible to go into all the variations of this latter plan, but there are two main divisions. In one there is a cut-out governor, and the power of the engine is regulated by cutting out the explosions altogether. In the other, the amount of oil is regulated, and there is an explosion every other revolution in all cases. The air is not throttled in either case. In most of the carburettors brought out in recent years for motor work the action is practically the same as in the earliest of the stationary engines. These had a spray carburettor, throttle control, and electric ignition. The arrangement for forming the spray is somewhat different, as it is usually a mixing valve of the type of fig. 74, or a float feed carburettor of ordinary design.

Whether these are likely to be more satisfactory now than they were years ago in stationary work remains to be seen. It is quite apparent that the problem of making an engine run satisfactorily on a car is far more difficult than that of making a satisfactory stationary one. The latter

always runs at one speed, and can have a cut-out governor. The former should be able to run at a great variety of speeds, and to run smoothly should be controlled by a throttle. This means that the compression and speed will both be constantly varying, and will have no relation to each other. On the other hand, there are several advantages. In the first place, low-tension electric ignition can be adopted. In the second, it may be an advantage to run the engine faster than was usual in the early days, and it may be that the older engines did not have enough compression to get the best results out of the spray carburettor.

In any case, it seems quite clear that to obtain satisfactory results the charge must be heated; therefore, the charge in the cylinder will be smaller, and the mean pressure lower than with petrol, so that an engine will not give as much power. Generally speaking, also, the best results with paraffin are obtained with a lower compression than with petrol.

Gas Producers.—Although, so far as I am aware, the suction gas producer has not yet been used on motor cars, there is a probable future for it in commercial motor traction, while in marine work firms are already turning their attention to it.

The principle of the suction gas producer is that air and steam are drawn through a mass of coal undergoing partial combustion. The whole of the carbon is burnt to CO, and the volatile products distilled off. The steam is decomposed into water gas—that is to say, the oxygen of the steam combines with the carbon of the coal, and the hydrogen is set free. The nitrogen of the air is unaltered. In the simple form used for small engines the suction of the engine itself draws the air through the producer, and there is no need of any other form of draught. The gas, on leaving the producer, must be cooled and cleaned from dust, tar, &c.

For actual details of the construction of stationary gas producers, the reader may be referred to works dealing with them, as they are beyond the scope of this work. In marine work the producer follows the lines of the stationary one with a few modifications.

Whether these can be so reduced in weight and volume as to be suitable for road work remains to be seen; it seems to be quite possible, as a similar reduction has been effected in steam engines, oil engines, and steam boilers, while lightness is the first consideration in road motors, and economy in stationary engines. The essential point in this case would be to make a satisfactory gas with the smallest possible producer. There seems no theoretical reason why it should not be made a good deal smaller than the stationary one and yet work satisfactorily, if economy is not the great object. The suction gas producer has very great advantages over the oil engine, if it can be made to work satisfactorily, and will probably come largely into use for marine work in consequence. It is not tied to the use of one particular brand of foreign fuel, but can use coal which is produced in England and a good many other countries. The fuel is also far cleaner to handle, as oil always oozes about and makes everything filthy, whereas coal is comparatively clean. Further, coal is very much cheaper.

The difficulties of making an engine that will give satisfactory combustion are far less than with oil, as the gas is a permanent gas and not a temporary vapour. Hence, there is no objectionable smell as there is with paraffin.

As compared with steam, the economy of the gas producer and engine in small sizes is very marked. Thus an ordinary steam engine will probably burn quite four times as much fuel as an ordinary gas engine worked with producer gas. Higher rates of compression are used with the latter than with either ordinary coal gas or petrol; but this may not involve any material modification of construction in the engine, taking the petrol engine as a standard. A powerful ignition is required, and the ignition plugs, valves, &c., must be easy to clean.

PART II.—CARS.

CHAPTER XII.

GENERAL ARRANGEMENTS.

Wheel Gauge.—In designing a car the first points to be settled are the width and length of the wheel base. The usual width varies between about 3 feet 6 inches and 5 feet in pleasure cars. Commercial vehicles are often made wider than this up to the limits permitted by statute.

The ordinary width of gauge for horse-drawn vehicles is from 4 feet 6 inches to 4 feet 9 inches, and as this is the result of long experience it is no doubt also the most suitable for motor cars. Many cars, especially the cheaper ones, are made much narrower, probably from considerations of expense and weight, but the saving is small, while the car is inconveniently cramped and jolts more.

Fig. 214 shows two cars of different gauges going over the same obstacle,

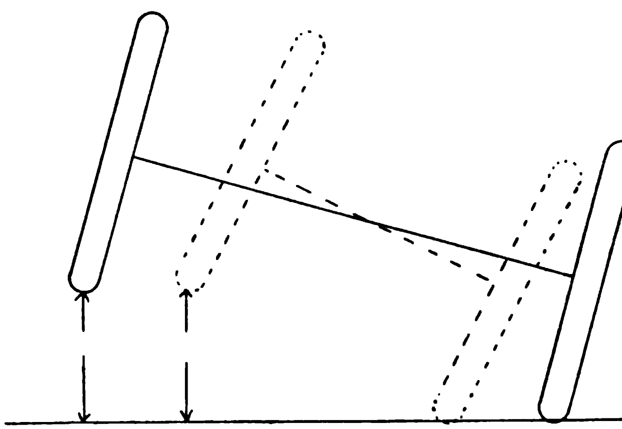


Fig. 214.

from which it will be seen that the narrower one is tilted over to a greater angle. The wider gauge, therefore, makes the car ride more easily, and also allows of the front wheels being put over to a larger angle without making the front of the frame unduly narrow; it is, therefore, easier to put in a straight frame, which is the strongest form. It also allows of the car being higher off the ground without being unstable.

Another much more important point is that the roads in England are most suitable for the standard gauge. Most of the less important roads

have two smooth tracks where the ordinary wheels run, which are often bordered by loose stones. If the roads are bad there are distinct wheel ruts. It is obvious that if the wheels fit in the smooth places the tyres will be subjected to less wear and damage. If the gauge of the car is the same as that of the ruts driving is easier, if it differs one wheel is always in a rut and the other just on the edge where the road is roughest.

This matter has received far less attention from English car makers than it has deserved, but, in America, practically all makers have always made all their cars, even the very cheapest and lightest, of standard track. It is largely because they have paid attention to this and other special requirements of their own country that they have retained the home trade in their own hands. All buyers, except that decreasing class who never go off a main road, should consider the fact that a car not of standard track will cost considerably more in tyres than one which is.

Wheel Base.—The length of wheel base depends upon what is to be placed within it. There should be little weight in advance of the front axle, and most cars now have the radiator just over this. The length taken up by the engine settles the position of the dashboard, and the length taken up by the body then determines the length of the frame. There is some choice as to the position of the back axle, but this is often settled by the body; for instance, if there is a side door to the back seats the hind wheels must be clear of it. If there is no side door the hind wheels may be a little in front of the back end of the frame, but this should not overhang so far as to make the car uncomfortable.

The actual length of the body will depend on its arrangement, and these will be discussed under the head of *Bodies.*

Briefly, there seem to be four distinct types wanted:—

1. *The Two-seater.*—This, generally speaking, requires a wheel base of about 5 feet 6 inches to 6 feet.

2. *The Small Tonneau.*—This type of car is really an extremely useful one, and has gone much more out of fashion than it deserved. It can be made on a wheel base very little longer than the two-seated car and with very little extra weight. Such a car of moderate power need not exceed 12 cwts., and is then probably the most useful car for those to whom cost of upkeep is of real importance and who can only keep one car.

3. *The Side-entrance Four-seater.*—This needs a wheel base of from 8 feet to 10 feet. This necessarily means a much heavier frame and more engine power to drive it; it is hence considerably heavier and more expensive than the foregoing.

4. Cars for more than four people may have any wheel base, up to that of a motor bus, which is about 15 feet.

Size of Wheels.—Carriages are usually built with wheels from 3 feet 6 inches to 5 feet or more; the average for ordinary work is about 4 feet 6 inches. Smaller wheels have often been tried, but are not satisfactory as they do not run smoothly enough. There are some difficulties in using wheels as big as this on motors; and the pneumatic tyres make it possible to have smaller wheels, but an increase in the usual size would be a great improvement. Even a very small difference in size makes a perceptible improvement, as I have found by experiment. When bicycles were first fitted with pneumatic tyres, wheels of all sizes were tried, and a good many machines were fitted with 26-inch wheels, but they were never successful owing to the extra vibration. Now for every different weight there

must be a size of wheel which is best for a given purpose. For racing, no doubt, small wheels are a great advantage as they can be made lighter, and the great question in a racing car is to get the greatest horse-power with a given weight. For comfort, however, things are different. When 28-inch wheels have been found necessary for comfort on a bicycle, a much larger one is evidently wanted on a car, yet many cars weighing half a ton have wheels only 28 or 30 inches in diameter. In addition to the extra vibration caused by these small wheels, they seem to be the principal cause of both tyre difficulties and side slip. The smaller the wheel the smaller is the part of it that touches the ground; consequently, the carrying of the car and driving it are both concentrated on a smaller piece of the tyre. In addition to this the wheel has to revolve oftener, so that not only is the tyre subjected to greater strains, but these are more frequent. In my opinion the smaller cars should have wheels not less than 34 inches, and all the larger cars 36 inches or more.

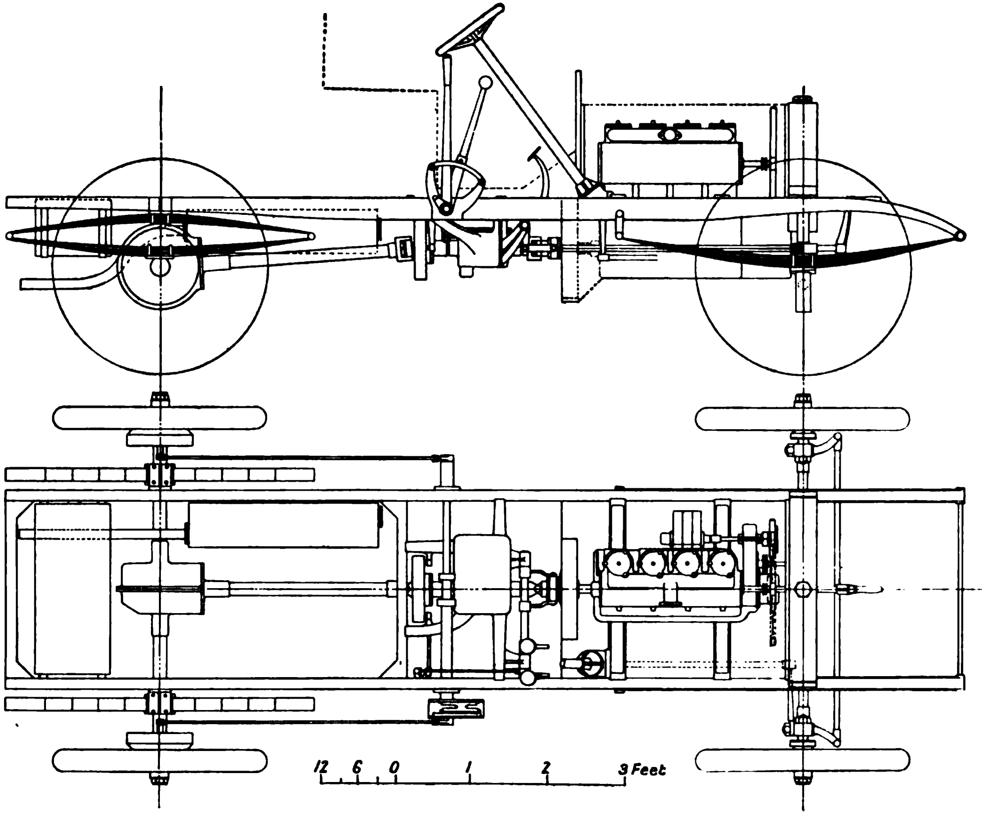
Anyone who follows the public trials of cars, such as the "thousand-mile trial" of 1903, &c., will see that the small cars fitted with little wheels have much more trouble with their tyres than the larger cars fitted with bigger wheels. This cannot be accounted for by the car being *lighter*, but must be attributed to the smallness of the wheels. In the "thousand-mile trial" the small cars had about seven times as many stoppages on the road due to faulty tyres, as the large ones.

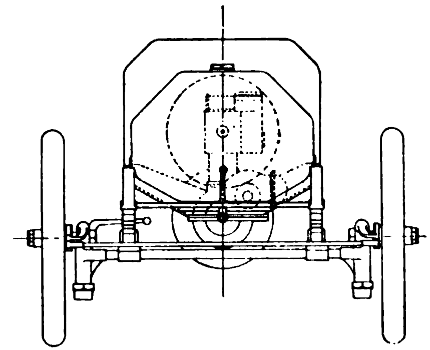
Large diameter tyres will, to a certain extent, compensate for the smallness of the wheels, but not at all effectively. Further, sideslip is then worse. The longer the contact with the road is in proportion to its breadth, the less the wheel will be likely to go sideways, instead of forwards. Further, the larger the wheel the easier will it surmount obstacles in a forward direction, and the less likely will it slide off them literally. As the small wheel has to go round faster, the wear on the bearings will be greater.

The question of fitting solid tyres to cars has often been discussed, and although the present tendency seems to be towards pneumatics for all purposes, there seems to be room for a good solid-tyred car for special purposes. But for this to be successful the car must be specially designed for solid tyres. In particular, the wheels must be of the size of the usual carriage wheels, say not less than 4 feet for the driving wheels. On the other hand, there is no reason why, with solid tyres, the steering and driving wheels should be the same size, as there is with pneumatic tyres. The front wheels might be smaller, as it is difficult to get enough lock for the steering with very big front wheels. With pneumatics it is desirable to have all the wheels the same size, as then the covers and air tubes are interchangeable and fewer spare parts need to be carried.

Luggage Room.—One point in connection with cars that remains to be satisfactorily arranged is to get room for luggage. In the ordinary horse-drawn vehicle there is the whole of the room under the seats, but in many cars the whole of this seems to be taken up with petrol tanks, tools, batteries, coils, &c. The result is that for all the ordinary work that carriages do one must use a four-seated car, to carry two persons only, and reserve the other space for the luggage.

In small cars the petrol tank can be put on the dashboard, and be moderate in size; this need not increase the length of the car as the feet can be put underneath the tank. In larger cars the tank is too large to be put



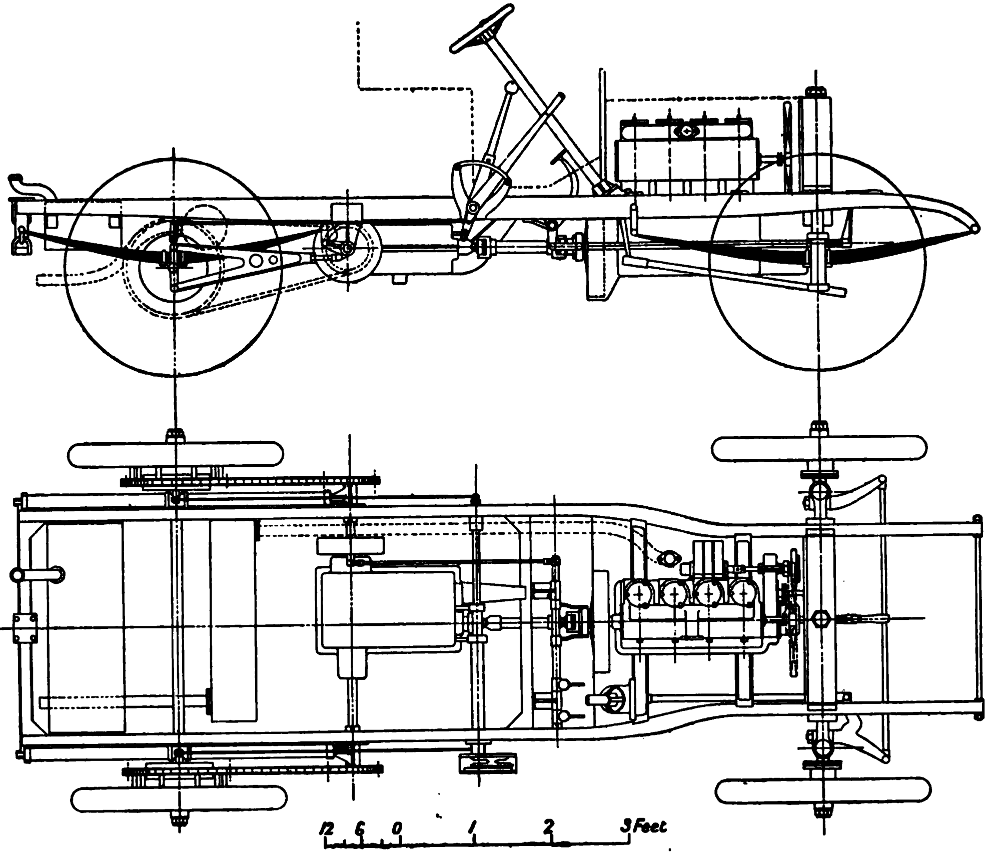


Particulars of Cars in Figs. 215 and 216.

Wheel base 8 feet 6 inches.
Wheel gauge 4 feet 6 inches.
Engine, 4 cylinders $4\frac{1}{4} \times 4\frac{1}{4}$ inches.
Three speeds and reverse change-speed gear-wheels 1 inch wide.
Pressure feed.
Engine, general arrangement as in fig. 22.

Fig. 215.

Back axle, generally as in fig. 261.
Crown wheel 12 inches diameter.
Gear box, general arrangement as in fig. 237.
Front springs, 4 feet 6 inches long; back, 4 feet long.



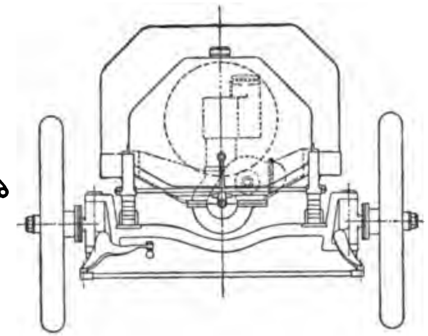


Fig. 216.

Gear box, general arrangement as in fig. 241.
Front springs, 4 feet 6 inches long.
Back springs:—Side springs, 4 feet long; cross spring, 3 feet 4 inches long.

there conveniently; a better place is inside the bonnet just behind the engine, but this increases the length of the car. Both these have very short petrol pipes, and the petrol may be made to run into the carburettor on any hill without difficulty. Many of the larger cars have the petrol tank hung underneath the chassis and a pressure feed is used. This seems the best place for it when too big to put on the dashboard, but it is a little more expensive owing to the pressure valve and pipe required; but it is sometimes very badly arranged, as it is hung so low as to make a most objectionable dust. The worst place would seem to be under the driver's seat, as here it takes up a great deal of room in the car and is inconvenient to fill. A still greater defect is that, unless the carburettor is placed very low, the petrol will not run into it fast enough when going up a steep hill. If it is placed very low it is almost inaccessible.

When small tools are wanted it should not be necessary for the passengers to get out of the car in order to obtain them. If there is a side entrance to the body the tools can be kept in a locker under the back seat opening from behind. The levers, &c., on the driving side should be so arranged that the driver can easily get out on that side. This is seldom the case although there is no great difficulty in managing it. Dust is of all others the greatest consideration at present, as speed largely depends on freedom from it. Some cars make much dust, even at a moderate speed, owing to the exhaust boxes, petrol tanks, &c., reaching too near the ground, and to the disproportionately large tyres on small wheels. If the car was raised farther from the ground there would be much less dust and the driver would be able to see better. Here, again, there is a great difference between the requirements of a racing and a touring car. The racing car is required to go round corners at the highest possible speed without capsizing. In the touring car, on the other hand, the speed depends on the extent of view, and with the high car this is greater than with the low one. There is absolutely no danger of capsizing even with the highest built car, going round corners at any rational or legitimate speed.

Figs. 215 and 216 show a general arrangement of a live-axle and chain-driven car respectively, which illustrate a good many points. Those points common to both cars will be noticed first, beginning at the forward end.

Front of Frame.—The frame shown has a cross-tie right at the front end; it forms the pins for the front springs, stiffens the front portion, and makes a very convenient place for carrying a head light, as the latter is well forward and out of the way of the starting handle. There is no need for a second cross-piece under the radiator, unless this is required to carry the front end of the engine; in fact, it is possible to do without any cross-pieces in the forward part of the frame, as the cross arms carrying the engine and gear boxes will generally be enough.

The radiator is generally dropped between the side frames, as shown, to get more surface without excessive height. Thus, if a cross-piece is put below, it is not very strong unless rather heavy.

Fan Arrangement.—There is generally a fan of some kind to suck the air through the radiator, and this may be either one driven by a belt, as shown, and placed just behind the radiator, or it may be in the flywheel. In the latter case, as the whole of the bonnet is made air-tight by the screen below the engine and flywheel, the air is sucked through; but to make it effective the flywheel must be rather large in diameter. An advantage of the fan in the flywheel is that the engine can be placed much

further forward and the bonnet shortened. As will be seen if the fan is placed between the cylinders and the radiator, it involves the former being some way behind the latter.

The length of the frame and wheel base will be determined by the body the car is intended to carry, that shown being suitable for a moderate-sized four-seated car. The wheel base may be lengthened or shortened as desired, the difference of length being between the gear box and back axle in fig. 215, and between clutch and gear box in fig. 216, all other parts remaining the same.

The petrol tank is in both cases shown under the back end of the car; to be used with a pressure feed. It is shown as being carried *inside* the frame, and, therefore, well off the ground. In many cases the tank is carried entirely below the frame and very close to the ground, the objections to which have been referred to. Taking the cars now in detail.

Live-Axle Car.—The gear box may be put as close to the engine as possible, after allowing room for the clutch. This provides the longest possible propeller shaft, and, therefore, minimises the work on the universal joints. The position of the change-speed levers is largely determined by the length of the gear box and the arrangement of the change-speed gear. If the gear box is short, as shown, the lever can be put behind the gear box and the brake and change-speed levers can be carried on the same bracket.

If the clutch or gear box is of such construction that the after end of the gear box comes further back, this arrangement is not possible as it would come too far behind the driving seat. The alternative is to put the change-speed lever quite in front of the gear box, but this usually throws it rather far forward and the gear box rather far back, or to lead the lever straight into the gear box. In the latter case it is seldom convenient to have the brake-lever on the same centre and quadrant as the change-speed lever, as the break-lever spindle has to go right across the car, and may interfere with the accessibility of the gear box. The dotted lines show approximately where the front seat will come, and in all arrangements of levers their position with regard to this must be considered.

The carrying of the gear box may be done in several ways. That shown allows of all the pedals being carried on the gear-box casting, so that there are no loose pieces to bolt on to either it or the frame; the brake is carried in the same way, the only loose piece being the bell crank lever to the brake. This will be a good deal cheaper in erecting than having a lot of loose pieces to bolt up; it is also lighter. The brakes are all shown as internal and interchangeable.

With an external foot brake the bell-crank lever is saved and the whole can be erected on the gear box before being taken into the erecting shop, but as external brakes are not suitable for the back brakes two patterns of brakes will be needed.

If the change-speed lever is taken into the side of the gear box the brake drum can be brought closer to the latter, and a cross arm used to carry it at the after end as well as the forward one. In this case, the bell crank for the brake can be carried on the casting, but a separate quadrant may then be wanted for the hand brake with the necessary attachments. Careful thought will be needed in these matters in order to secure convenience and to avoid unnecessary expense.

The propeller shaft may have either one universal joint, as shown, or two, as in fig. 6. In the former case the tube in which the propeller shaft

runs forms the radius rod. This may, or may not, be supplemented by side radius rods, as thought fit (see Chapter xiv.). In the arrangement shown the front end of the radius rod is held rigidly on the cross piece of the frame and no other radius rod is provided.

Chain-driven Car.—The arrangement of the engine and parts in front of the dashboard is the same as in the live-axle car. The great difference is in that of the gear box. When the differential is contained in the change-speed gear box this must be placed a great deal further back in the car than in the case of the live axle one in order that the chains may be short. This brings the gear box well behind the change-speed lever, as a rule, so that the arrangement will be much as shown. If the car has a very long wheel base the gear box will be still further back. The result of this is that the gear box is entirely underneath the body of the car and must be so arranged that the parts can be reached from the bottom. As the gear box is so far from the engine it is not possible to carry the pedals on it, and, therefore, either the pedal shaft must go right across the car or there must be a cross piece to carry it, as shown.

The different methods of carrying a gear box for a chain-driven car are mentioned in the chapter on transmission. As shown, the weight is carried on the cross shafts behind and on the cross piece of the frame in front.

It is possible to have the gear box, &c., all arranged as in a live-axle car and to have the cross shaft in a separate casing like a live axle. This is shown in fig. 5. The advantages and disadvantages are noticed in Chapter xiii.

In fig. 215 the frame represented is straight, while in fig. 216 it is widened out behind the dashboard. The latter is the more usual plan, but the former could be made quite as satisfactory if the body of the car was built wider than the frame. If the frame is widened behind, its full width should be at the dashboard, or it will unduly narrow the floor of the front seats.

For purposes of comparison the cars are shown with different methods of springing, and also with different types of front axle. The springs are in both cases somewhat longer than are often fitted, but probably not too long for ordinary use on English roads. As will be seen, there is no difficulty in fitting them in.

In the type shown in fig. 216 the steering gear comes so near the ground that the rod from the steering gear to the steering arm on the front axle has a downward slope. The steering would consequently be disturbed by the motion of the springs. To avoid this the rod should be in a line with the front end of the spring. In practice, the disturbance from this cause is perceptible, but not very great. It is also very objectionable to have the rod connecting the steering arms so close to the ground, as, if damaged by obstruction, it may cause a serious accident. If there is no inside frame the steering gear must either be fixed to the side frame or to a cross piece. The frame is usually so wide that to bolt the steering gear to it would bring it too far out of centre of car, so it is shown fixed to one of the engine arms where it can be bolted to a machined facing.

In both cars the change speed is on the gate plan with three speeds and reverse.

CHAPTER XIII.

CLUTCHES—TRANSMISSION—DIFFERENTIAL GEARS—UNIVERSAL JOINTS.

Clutches.—The connection between the flywheel of the engine and the gear box is made by a friction clutch, which has several features not usual in friction clutches. The most usual friction coupling, in most classes of machinery, is a belt running on fast and loose pulleys, but there is not room in a car for this to be satisfactory.

The special requirements of a clutch for a car are :—

1. A very small rotating momentum, so as not to injure the gear in changing.
2. Absolutely free when out of gear.
3. Very simple to repair and easily understood.
4. It must be possible to let it slip for some considerable time without injury, transmitting part of the power all the time.

In addition to this it is often desirable that it should be as compact as possible lengthways, and in all cases cheapness of construction is a great advantage.

The three most usual forms of clutch are :—

1. The cone.
2. The expanding.
3. The multiple disc.

Of these the *cone* clutch is the oldest, simplest, cheapest, and the easiest to understand. It is a very simple casting, and can be made of aluminium and be light at the circumference, so as to have little momentum. The *expanding* clutch is much more complicated. The cone clutch consists of but one sliding part and a spring; the expanding clutch of many pieces; consequently, the latter is much more expensive. It is also much heavier for the same diameter, as there must be some arrangement in the circumference for expanding it. Consequently, it is generally made of very small diameter in order to keep the momentum down. This means that the pressure between the surfaces must be very great in order that the clutch should transmit the power required, and the movement of the parts small in order that the foot pedal should be capable of withdrawing the clutch without serious effort. This means a delicate adjustment to allow of their holding firmly when in action and being quite free when out of gear. Its use is not very extensive now and is likely to be less in the future.

The *multiple disc* clutch has many advantages. Its action may be likened to that of several cone clutches on the same shaft actuated by the same spring. Sometimes the discs are cones, but in others they are flat; either form is about equally effective.

The advantages of the disc clutch are :—

1. That the surface is very large in proportion to the power transmitted, and there is, therefore, little or no jerking.

2. That the same spring acts on all the discs, and, therefore, the clutch can be made to take up a very large power with a very moderate spring. This avoids all difficulties in taking the thrust of the spring.

In comparing these clutches the plain cone is the simplest and cheapest, and probably for small cars the best. On the other hand, it does not take up the power so easily in the larger cars, as the diameter is limited by the size of the flywheel, while the spring should not be stronger than can be conveniently worked by the foot.

Some other form of clutch will, therefore, be preferable for the larger cars. In comparing the multiple disc and the expanding clutch the advantage seems to be with the former. It is generally cheaper, has more margin of surface, and is less complicated, while the revolving parts are lighter and, therefore, easier on the gear. Nevertheless, clutches made on each of these plans have been successful.

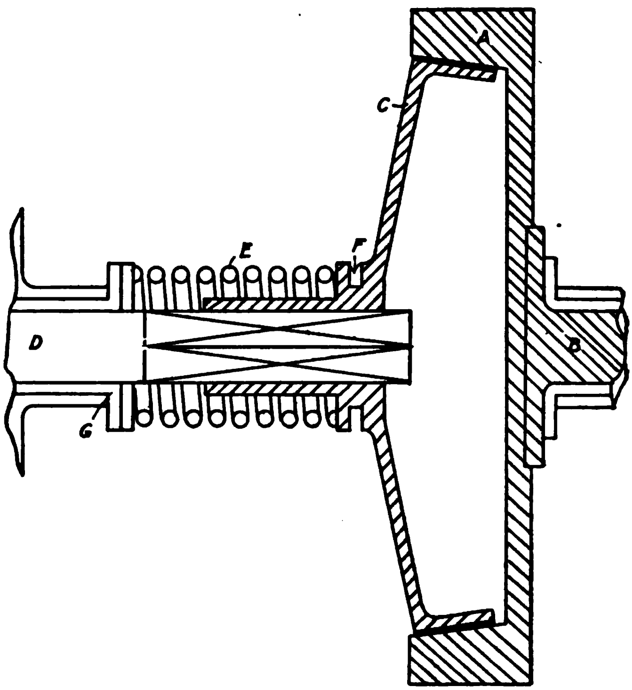


Fig. 217.

A, Flywheel.
B, Crank shaft.
C, Cone clutch.
D, Shaft from gear box.
E, Spring.
F, Collar to withdraw clutch.
G, Bearing at end of gear box.

Cone Clutches.—Fig. 217 shows the primitive form; fig. 218 a slight modification, in which the thrust of the spring is taken on the shaft itself and, therefore, is not communicated to any of the other bearings when the clutch is in gear.

Either of these forms has it disadvantages. One defect is that the

driving part will not slide easily on a square shaft unless well oiled, and hence the pressure cannot always be regulated with the desired nicety. A key is better and cheaper, but not at all perfect. Another defect is that as the clutch is entirely carried on the gear box it must be absolutely in line with the engine in order to become quite free. Further, if the gear box gets the least out of line with the engine the side pressure on the bearings causes wear.

The latter defect can be avoided by having the frame so rigid that it does not twist, and by carefully lining up the gear box to the engine, but it is perhaps a better plan to have a clutch that is not so dependent on this. Fig. 219 shows a plan in which the clutch is carried entirely on the engine and drives the gear box through a claw clutch. The only disadvantage of this is that there is a slightly greater tendency of the clutch to spin when taken out of gear, owing to the friction; but this is not found serious in

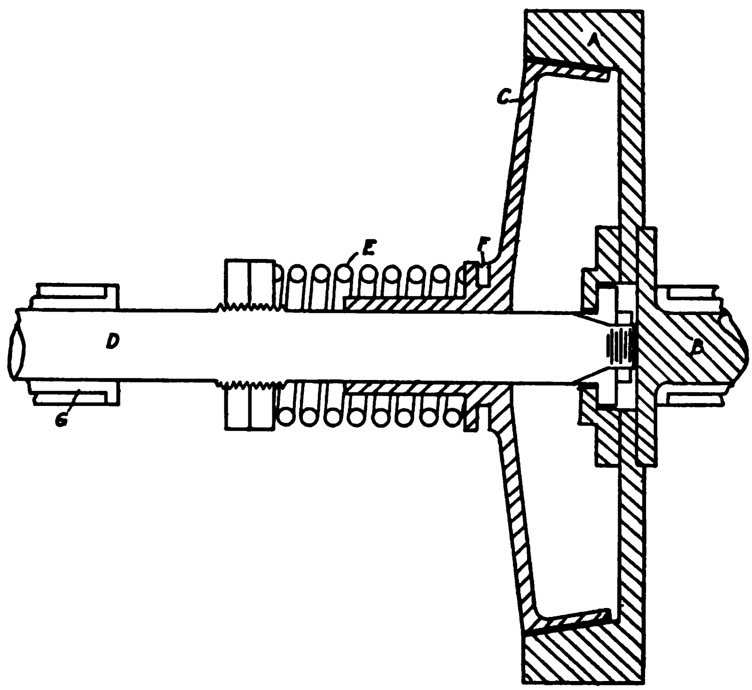


Fig. 218.

practice, and can be avoided by fitting a stop for the clutch to press against when the pedal is pressed down to its extreme limit.

In figs. 217 and 218 the spring is shown concentric with the shaft, but in fig. 219 it is outside. The latter has many advantages, the only objection being that the thrust of the spring is always taken through the collar. On the other hand, the spring is much easier to get at, while the clutch can be made easily dismountable if the gear box is close to the engine. It is, of course, quite easy, in this pattern, to carry out the spindle and put

the spring on it; or, in the case of figs. 217 and 218, to put the spring on the pedal.

If the gear box is a long way behind the engine the clutch can easily be made to take down without removing any part of the gear box, &c. If the gear box is close up to the engine this requires more care. It should be possible to take the clutch entirely out of the car without disturbing anything except its immediate connections; the simplest way of doing this is to make it in two halves, bolted together. Sometimes there is a short length of shaft between the clutch and gear case which can be taken down to allow

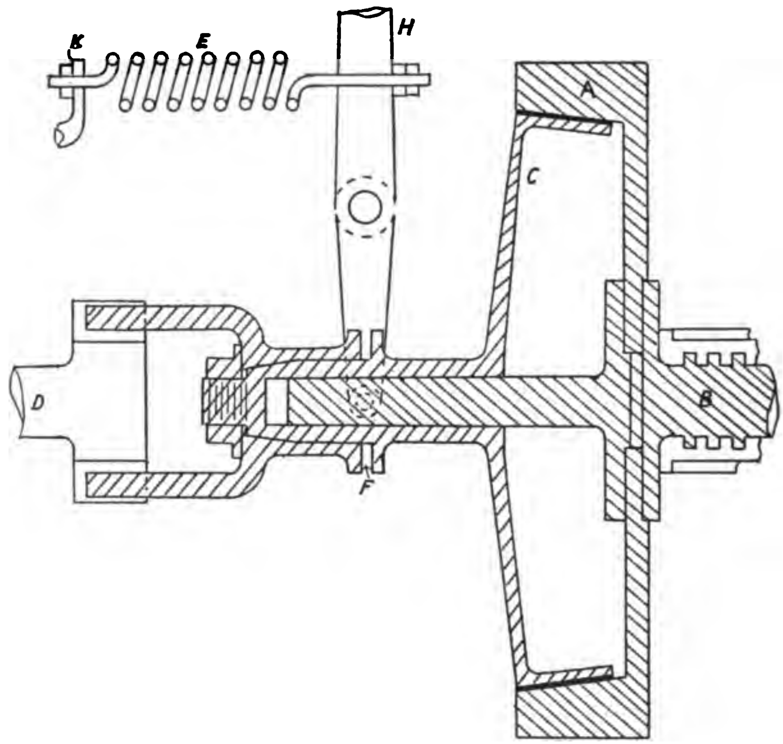


Fig. 219.

Letters as in fig. 217.
H, Foot pedal.
K, Attachment of spring to frame of car or gear box.

of the clutch being threaded off the spindle it runs on. This is perhaps a little cheaper, but puts the gear box further back. With a clutch of the type of fig. 219 the bolts which secure the stem on which the clutch runs to the crank shaft can easily be made to undo without disconnecting anything else, and the whole dropped out. There are many variations in the cone clutch, some of them very complicated, but they have no apparent advantage over the simpler arrangements.

An inverted cone clutch, as in fig. 220, is sometimes used, in which the thrust is entirely self-contained, but the spring is not easy to get at. The

loose ring of the flywheel being bolted on in two halves makes the clutch easy to get at, but adds considerably to the expense. It makes a clutch which is very short indeed lengthways. With modern push pedals it is not, as a rule, so easy to arrange the pedal gear in this as in the plain cone clutch.

The periphery of all cone clutches should be as light as it can be made; hence it is generally an aluminium casting bolted on to a steel centre. Possibly a still lighter clutch could be made of stamped steel plate.

The angle of the cone has the most important effect on the satisfactory working of the clutch. If the cone is too steep it is impossible to make the clutch take up gently, and either it grips hard or it does not grip at all. On the other hand, if too flat a great deal of pressure is required to make it grip at all. A taper of about 1 in 8 is usual.

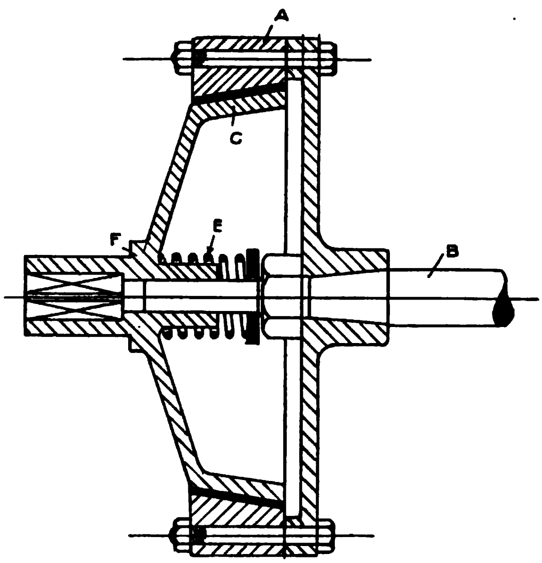


Fig. 220.

Several variations of the ordinary cone have been made in order to make it take up more easily. One is to have two clutches of different sizes so arranged that the smaller one comes into operation first, and the larger one after. This is slightly more expensive, and rather heavy compared with the simple cone. The leather has also had springs placed behind it to make it engage more easily. A properly proportioned cone clutch of the simple type seems, however, to meet all requirements for engines up to, say, about 20 horse-power.

Cone clutches are covered with leather. Metal to metal cone clutches running in oil have been used, but not to any great extent.

Expanding Clutches.—Various designs are shown diagrammatically in figs. 221 to 223. All consist of many pieces, which renders them expensive and delicate. One weak feature is that the centrifugal force tends to expand the clutch, and, therefore, the faster it runs the tighter it holds.

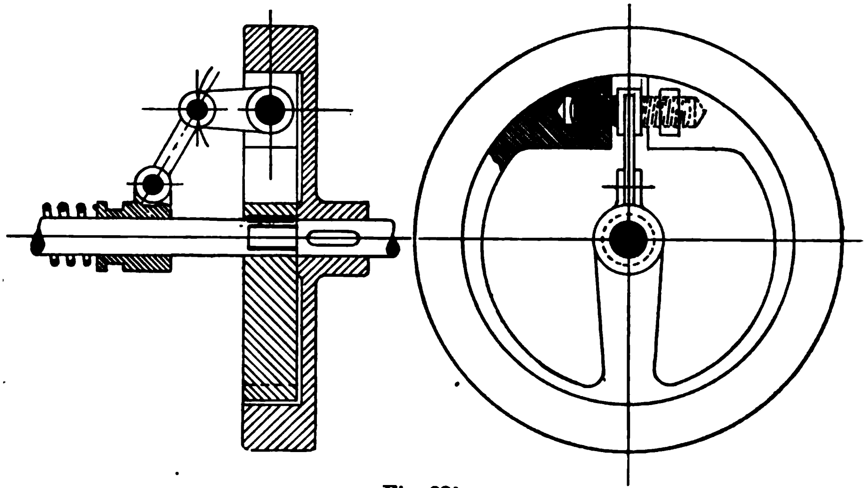


Fig. 221.

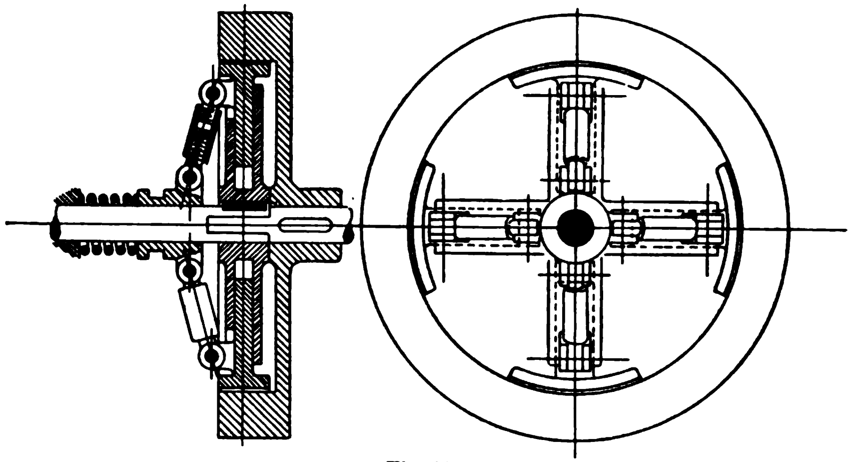


Fig. 222.

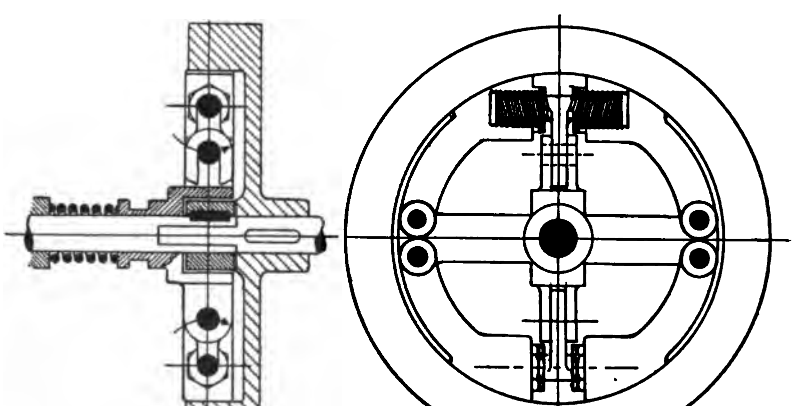


Fig. 223.

For this reason all expanding clutches should have some positive means of withdrawal, and this must be powerful enough to overcome the centrifugal force without undue effort.

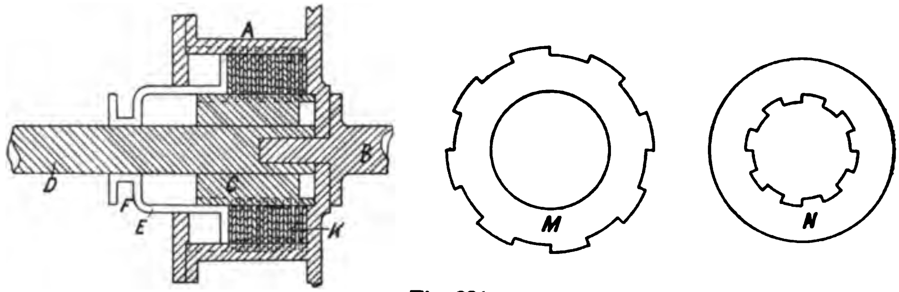


Fig. 224.

A, Casing of clutch carried on flywheel.
B, Crank shaft.
C, Sleeve fixed to shaft D.
D, Shaft from gear box.
E, Sliding sleeve with spring to press against discs, K.
F, Collar to withdraw clutch.
K, Friction discs alternating as M and N.

A and C are slotted to take the slots in M and N respectively.
On E pressing all the discs, K, together their friction drives C.

In the clutches of the type shown in figs. 221 and 222 the action is that of a toggle joint—that is, the clutch starts with a quick movement and finishes with a slow, powerful one. This gives a very powerful grip with a light spring, but the clutch must be very accurately adjusted to work properly.

Multiple Disc Clutches.—For high powers the best clutch would seem to be the multiple disc (fig. 224). It consists of a casing in the flywheel with longitudinal keyways on its inside; a shaft, also with keyways; and a series of discs with slots fitting alternately on the shaft and inside the casing. Each alternate disc rotates with the casing and shaft respectively, so that when all are pressed together the friction between them transmits the power. The casing is amply supplied with oil, and as the discs have slots cut in them through which the oil can circulate, they are always lubricated. The usual number of discs in this type of clutch is from 20 to 60 and the diameter about 6 inches.

Discs are often made of the shape which was introduced by Professor Hele-Shaw, and is somewhat as fig. 225. They are steel stampings and really form a series of cone clutches. The stampings are somewhat lighter than the flat discs and need not be machined.

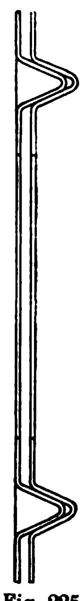


Fig. 225.

Section of one pair of Hele-Shaw discs.

Both these forms of disc clutch work extremely well. Their only real disadvantages are that they are somewhat dependent on the thickness of oil used for their regular action, and they are much more expensive than the cone clutch. As in the case of the cone clutch, the spring controlling them may be mounted on the shaft or on the pedal. The latter seems the best, as the pressure to be transmitted

is very small, and it is a great advantage to be able to easily adjust the spring. Also, it saves a certain amount of space. It is not quite so easy to arrange for the clutch to be taken down without disturbing the gear box with the multiple disc as with the cone clutch, but can be so arranged with care.

In all cases where a type of clutch is employed which transmits any thrust to the engine, provision must be made for taking it. Fig. 219 shows the ordinary form of thrust bearing as used in marine work, but so elaborate an arrangement is hardly necessary for the slight thrust of a clutch, since a simple flange on the after bearing, as in fig. 217, seems perfectly satisfactory, and cheaper.

Ball thrusts are occasionally used, but for taking the thrust on the crank shaft are unnecessary, as the thrust can be taken on a well-lubricated bearing. They are not very easy to get on, as the flywheel is usually fastened on with a flange, and, in this case, the thrust ring cannot conveniently be got past the flange. On the other hand, for all the thrust bearings outside the crank case, ball bearings seem to be far the best, as they will run well without lubrication. For instance, in fig. 217 there should be a ball thrust at each end of the spring. In figs. 218 and 220 the thrust is self-contained when the clutch is in gear, but it is as well to have a ball thrust to take the thrust there is between the crank shaft and the clutch when the latter is out of gear. Otherwise the clutch is liable to go on spinning when out of gear from the friction of the thrust. In fig. 219 there should be a ball thrust to the collar, as this always takes the thrust of the spring, but, in other cases, there is no great advantage, as the thrust is only on the collar when the clutch is out of gear. If there is no ball thrust, the collar may be put on in two halves, and this is often convenient, as the groove for it can be turned out of the solid, but, if there is a ball thrust, there must be some arrangement by which the thrust and collar can be put on in one piece, as in fig. 219.

Transmission.—The two main divisions into which the transmission gear of cars fall are those of the "live axle" and the "chain drive." It is true that there are cars which do not have a live axle and yet are driven by gear, and there are also cars which are driven by chains and yet have a live axle. These are, however, usually of special types, and do not have the engine in front in the usual way. Figs. 215 and 216 show the two systems. It will be seen that in each there is the engine in front. There is a friction clutch driving a change-speed gear box. There is a bevel gear and differential. In the live axle type, however, the bevel and differential are on the back axle, while, in the chain-driven type, these are on a countershaft, and the drive from this is by separate chains to each wheel on the back axle. In the case of the live axle, the drive is taken by a shaft with universal joints from the gear box to the back bevel gear case.

The advantages of the live axle type are principally:—that the chains, which are always more or less noisy and subject to wear, are dispensed with. All moving parts are cased in. The gear box can conveniently be put well forward where it can be inspected by taking up the floor boards in front of the driving seat, while the only alteration in parts to make chassis of different lengths is a different length propeller shaft. There is a considerable saving in expense, as the universal joints do not cost anything like as much as the chains and back axle, and all other parts are common to both. Further, the loss of power from driving through the chains is avoided, though this is probably small as long as the chains are new.

The chain drive probably originated with the desire to use the ordinary carriage axle, and has remained in use rather as a matter of custom ever since. Still, it has certain undoubted advantages for some classes of work. One of these is, that it permits of a much greater ratio of reduction between the engine and road wheels. Whether a direct drive is used or not in the gear box, it is not convenient to gear the top speed down materially here, or the ratio of the wheels in the lower speeds becomes very great, nor to make the ratio of the bevel gear more than 4 to 1, or the gear will not run quietly. Consequently, for a ratio of more than 4 to 1 on the top speed, there must be a countershaft in addition to the live axle, which means noise and loss of power. On the other hand, with the chains there can be a reduction on the chain wheels of 2 to 1 or more, and they will work quietly. This means that, with a bevel ratio of 4 to 1, an 8 to 1 reduction is possible between the engine and road wheels. This point is not of any great importance in ordinary pleasure cars, as they are now generally provided with enough engine power to drive at least 3 to 1 on the top gear, and many of them will even drive 2 to 1. This means that the engine can be run a great deal slower. In the future it seems probable that cars will be given still more power in proportion to their weight, so that a 3 to 1 gear will be the highest ratio likely to be required.

Besides this, an advantage for purely high-speed work has been that the height of the car is entirely independent of the height of the wheel centres. In the live axle type the floor of the car must clear the casing of the back bevel. This I do not regard as the least objection in an ordinary touring car, as it is not convenient to be too low down. There will, however, be special uses for which it will be suitable.

If the car is considerably geared down on the chains, the strains on the cross shaft and differential are diminished, and they can be made lighter. Pleasure cars, however, now have the chain wheels nearly the same size, so as to make the reduction here very small. Where there is much reduction the shaft will, of course, run faster, and therefore, although there will be less strain, there will be more wear. For this and other reasons it is not good to have the reduction on the chains too great, and it should probably not, in any case, very much exceed 2 to 1.

There are two claims often made for the chains that it is well to dispose of. The one is that they are "elastic." Now, it is the whole object of the chain-maker to make his chains with absolutely no stretch. As a matter of fact, in all chains that are used there is no perceptible stretch. The chains may be slack, and, in fact, there must be a certain amount of slack in them, but this does not "reduce the shocks" on the machinery as is sometimes claimed. On the contrary, it produces shocks from the fact that it allows of motion before the strain comes on the parts, so that when it does there is a jerk. If it is desired to have a spring drive between the gear box and the wheels it can be easily arranged in either kind of drive, but there does not seem to be either necessity or advantage in it. The other claim is perhaps hardly worth alluding to. It has been persistently stated by some advocates of chains that there is doubt as to whether the live axle can be made strong enough to carry a heavy car. As all locomotives, railway carriages, tram cars, traction engines, and nearly all lorries are carried on live axles, there is no room for any such doubt.

It is rather curious that the live axle should have been preferred for the small cars, whereas the chain drive has been largely used for the larger ones.

From a mechanical point of view, the advantages of the live axle are more prominent in the larger sizes, and its disadvantages are less. As has been shown, it is not easy to make the bevel gear run quietly when the reduction is more than four to one. Now, in many of the smaller live-axle cars the reduction has to be something like 6 to 1 to give them enough power to take their top speed properly. In this case the bevel gear is noisy, but by having chains the bevel gear can be reduced to reasonable proportions. Further, chains have been more successful in light drives than heavy ones. On the other hand, the larger cars are always geared a good deal higher, as they have plenty of engine power. Probably the days of the very low-powered car are nearly over, and as there will be very few cars which will not take 3 to 1 gear this objection will not apply.

The possible variations of the arrangement of the actual change-speed gear in the gear box are very numerous. Most cars are made with either three or four speeds. The great division in arrangement is that some have a direct drive on one speed, usually the top, and others have a shaft-to-shaft

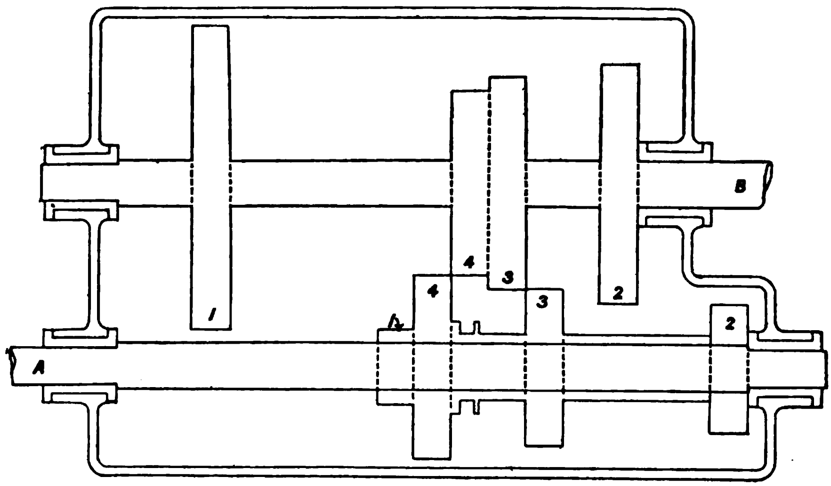


Fig. 226.

In figs. 226 to 234, A is shaft from clutch; B, shaft to back axle or chain wheels. Gear wheels are numbered for the speeds, and R is reverse. All are to the same scale for gear wheels of the same width and centres, and in all cases are shown without clearances between teeth.

arrangement for all speeds. The shaft-to-shaft plan is the older method, while the direct drive has only lately come into general use.

Fig. 226 is a diagrammatic view of a gear box with four speeds and a shaft-to-shaft drive. In this the frictional loss is the same on each speed, as it is the amount lost in one pair of wheels. Fig. 227, on the other hand, shows a gear box arranged for a direct drive on the top speed. In this case, when the car is running on the top speed, the back shaft is running idle, and the drive does not go through any gears at all. There is, therefore, practically no loss. On the other hand, when on the other three speeds there is *twice* the loss there is in fig. 226, as the drive is through *two* pairs

of gear wheels. This must not be forgotten in settling which type of gear to have, as it means that it is not worth while having a direct drive on one speed, unless it is used more often than all the other speeds put together.

If a bevel gear is enclosed with the gear box in a chain-driven car another variation (fig. 232) has some of the merits of both plans. In this there are two bevels, one on the direct shaft and the other on the counter-shaft, so that while there is a direct drive on one gear, there is a shaft-to-shaft one only on the others. Thus, in this case there is the same efficiency as the direct drive on the direct speed, while there is only the same loss on the others as in the shaft-to-shaft. Thus, theoretically, it combines the advantages of both types. In practice there are some disadvantages in construction, which will be referred to later, and it is not a type that can be applied to the live axle, as it would require two universally-jointed shafts, or else that the change-speed gear should be on the back axle, both of which are rather impracticable.

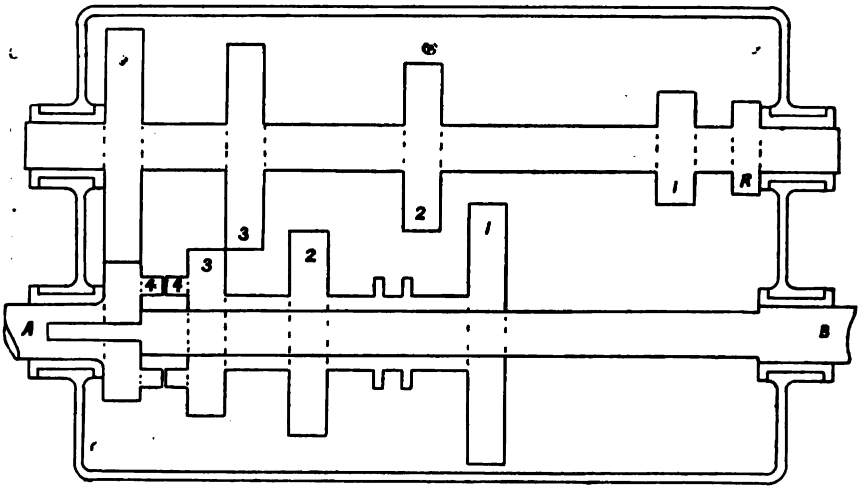


Fig. 227.

Comparative Efficiencies.—In order to compare the efficiencies of these various types, the ratio of loss from each pair of gear wheels, and also from the chains, must be known; but the data for this are difficult to obtain. In fact, it probably varies a good deal under different circumstances. It seems probable, however, that the loss is about 10 per cent. in each pair of gear wheels, and 5 per cent. for a chain, when the latter is new and in good condition. It will be understood that this is assumed; but it is one that seems fairly to fit the observed facts of motor performance, and it is, at all events, good enough to take as a basis of comparison between the different types of drives.

In all cases the transmission will be through a bevel gear, and in the chain-driven car the side chains in addition. For the ordinary types of drive efficiencies are as follows. They are given for a four-speed car with the direct drive on the top, if there is one. The last type, fig. 232, is called "Mors" type, as it was introduced by the maker so named:—

No. of Speed, . . .	1st.	2nd.	3rd.	4th.
	Per cent.	Per cent.	Per cent.	Per cent.
Live axle direct on top, . . .	70	70	70	90
Live axle shaft-to-shaft, . . .	80	80	80	80
Chain drive direct on top, . .	65	65	65	85
Chain drive shaft-to-shaft, . .	75	75	75	75
Chain drive, Mors pattern, . .	75	75	75	85

These values are believed to be sufficiently accurate for the purpose of comparison.

Consideration of number of Speeds.—In considering the number of speeds required, the fact that a direct drive diminishes the efficiency of the others must be borne in mind. If there are four speeds the ratio is usually 1, 2, 3, 4, or thereabouts. If the tractive forces of these are worked out it will be found that the hill-climbing power of the third speed is little above that of the fourth. It is 25 per cent. lower and has 20 per cent. more friction. In actual practice this is found to be the case, and my experience is that, with a direct top speed, the next lower must be 33 per cent. less in order to give an appreciable increase of hill-climbing power. This being so it seems as if with a direct top speed it is hardly worth while having four speeds. If there are fewer speeds, a considerably larger engine may be purchased for the same money, and the car will be more generally useful. In discussing the matter it is often forgotten that it costs very little indeed to make an engine a bit bigger if everything else is the same, and that this will give a far greater increase of average speed than an extra change of speed will. Besides this, the bigger engine will give a much quieter running car, as more of the hills can be taken on the direct top speed, and the engine need not so often be run up to its full power. My own idea is that the top speed should generally be used for running the car, which should take all ordinary hills on this speed; that the second speed be used for really steep hills, and that the first be reserved for special occasions. The latter speed should be low enough to be absolutely certain of going up anything that the wheels will grip on, even when the engine is not running its best or possibly when one cylinder is not working.

It has been suggested that though three speeds are enough when the top is direct it would be an advantage to have a speed above this that is not direct. This would only be used when running slightly down hill, or with a wind behind, and "would avoid racing the engine" under these circumstances. The same objection applies to this as to the four speeds in the other case. It increases the cost, and the money would be better spent in putting a bigger engine in and raising the gear all round. It is quite possible to make a car which will go any speed up to and over 60 miles an hour, and take ordinary hills on its top speed, without in any way injuring its engine, if the latter is big enough.

The custom of giving the large cars with powerful engines four speeds and the small low-powered cars only three or even two, is absurd, as, in the latter case, it is necessary to get the best result out of the engines and, therefore, more speeds are required.

A two-speed car with a very large engine may possibly prove to be the best for general use. Such a car would not, perhaps, be very fast on the

flat, but would well maintain its speed up the hills and would start quickly. This will suffice if a good average speed is the desideratum.

In all cases it must be remembered that in pleasure cars the most important points are durability and quietness, which will be secured by the general use of the direct speed and driving as little as possible through many shafts. For some purposes it will be better to keep a small engine and run it always at its best power; in this case many changes of speed will be wanted, and probably a shaft-to-shaft drive will be the best. Extreme economy of fuel is of little importance in a car compared with comfort, as the total fuel bill for the year is very small, but in commercial work it is different.

Gears.—There is a great diversity of ratios between the gears, and the best will not be the same for all work. Roughly speaking, it is convenient to make them with a calculated tractive force of about 600 lbs. per ton on the first, 300 lbs. on the second, and 200 lbs. on the third in a three-speed car with a direct top speed (see Tables at the end of the volume for actual ratios).

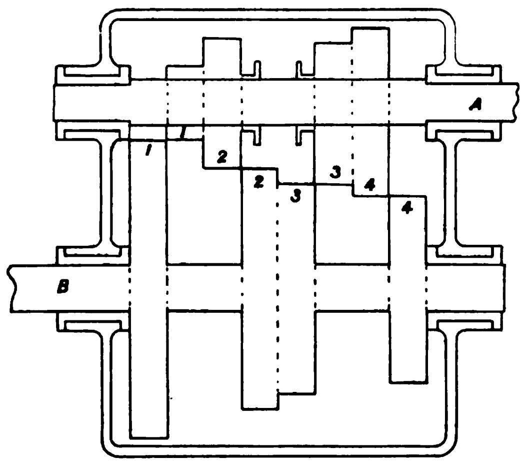


Fig. 228.

In the actual construction of the gear boxes and arrangement of gears there are many variations. The older plan was to have all the gears on one sliding sleeve, as in fig. 226. This may be called the "run-through" arrangement. Now, however, it is common to have the gears in two parts, so that these are moved independently, as in fig. 228; this may be called the "gate" plan. Both these figures show the arrangement for a four-speed gear box with shaft-to-shaft drive. It will be seen that the gate plan shortens the gear box enormously. This reduces the weight of the box without making the gears smaller. The gate plan also allows of the shafts in the gear box being considerably shortened, which helps to keep the gear quiet, as there is less chance of their springing and thereby throwing the gears out of pitch.

If a shaft-to-shaft drive is used, it is not of very much importance whether the sliding gears are on the driving or the driven shaft. It is, in fact, rather a matter in which the design is made to suit the circumstances. The main point is to so arrange that the wheels are easily renewable and

reasonably cheap to make. Having the sliding part on the driving shaft makes it lighter, as it is the smaller wheels that slide. It also makes the square shaft on which they slide the one with the least strain on it, at least

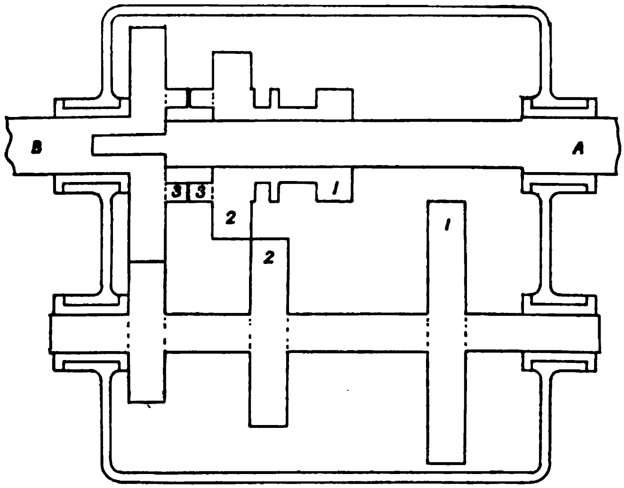


Fig. 229.

in theory. On the other hand, it is probably easier, when the driven wheels are the sliding ones, to make each wheel a separate piece, and it is easier to arrange the reverse. This latter point will be dealt with later.

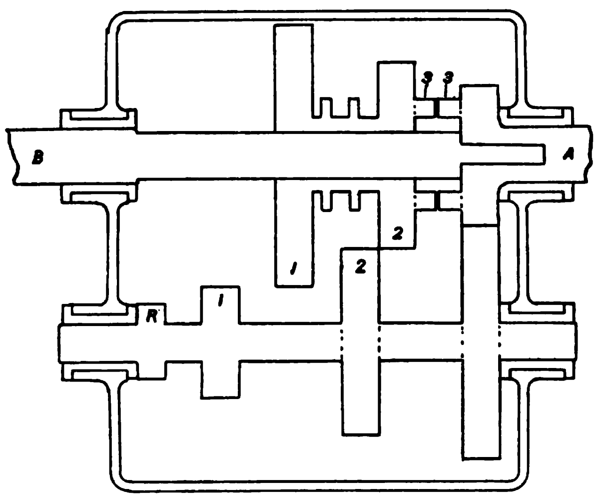


Fig. 230.

On the other hand, if the gear box has a direct drive there seem to be very distinct advantages in making the sliding gears on the driven shaft. Figs. 229 and 230 show two gear boxes in diagram with three speeds and

reverse, the one having the sliders on the driving shaft, and the other on the driven. The ratio of speeds is the same in each case, and differ little from that commonly adopted. For the lower speeds there is not much to choose between them. In each case the speed is geared down more or less to the countershaft, and then geared down again to the propeller shaft. On the top speed, however, there is a great difference; as in the one case the countershaft is driven considerably *faster* than the engine, and in the other considerably *slower*. This is unavoidable, as, if we make the ratio of gear between the countershaft and propeller shaft such that it runs at the engine speed or less on the top speed, the ratios between the engine shaft and countershaft at the low speed will be very great. In fig. 229 the back shaft runs 1·7 times as fast as the engine in the top speed, while in 230 it runs under two-thirds the engine speed, consequently in 229 it runs about three times as fast as in 230, and there is proportionately more noise and wear. Further, if the countershaft is driven slower than the engine there is no objection to having a

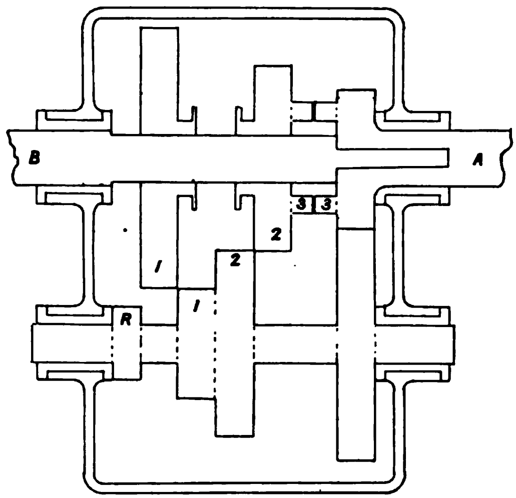


Fig. 231.

reverse wheel always in mesh with the countershaft, and always running. It can be made to run a good deal slower than the countershaft, and therefore so slow that the friction and wear are negligible. On the other hand, if the countershaft is driven *faster* than the engine, the reverse pinion would have to be driven very much faster than this again, or would have to be of impracticable size. Consequently, the reverse gear is always arranged to go entirely out of gear when not in use in cars in which the sliding parts are on the driving shaft, and this generally adds to the expense and complication.

In theory when the slider is on the driven shaft this should have more strain on it than when it is on the driving. In practice, however, there does not seem to be anything in this, as the shaft is generally as big in the latter case as in the former. This is because the strains are largely due to inertia.

If the gear box has a direct speed the gate plan of changing can be used equally well as with the shaft-to-shaft drive. Fig. 231 shows one with three

speeds and reverse. In this case the two halves of the gear each take two positions. One is the top and second; the other the first and reverse. With a four-speed gear and reverse the simplest way is to place the reverse beyond

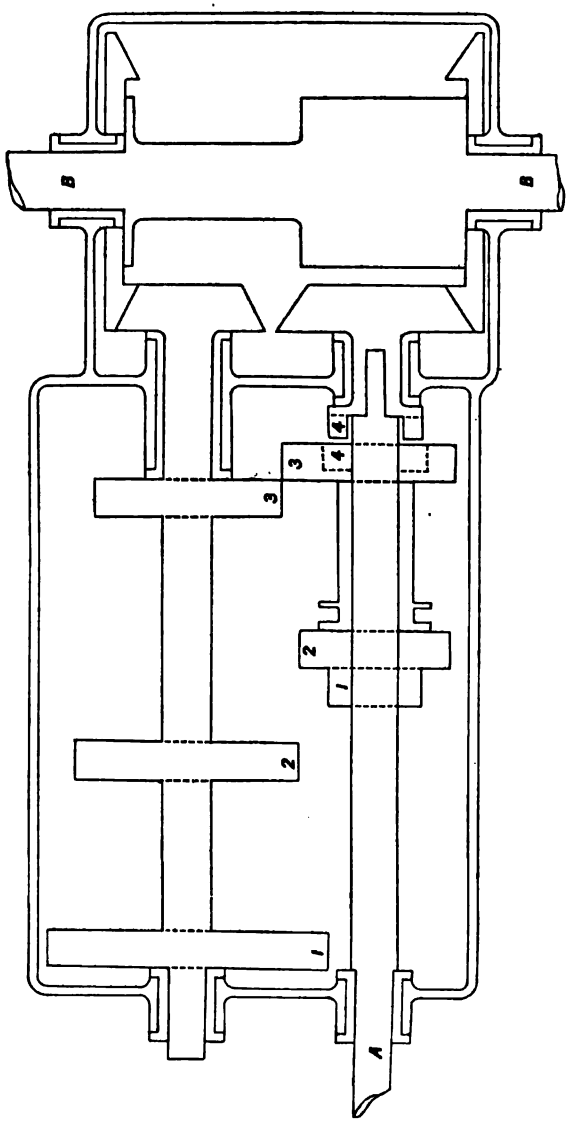


Fig. 232.

the low speed, so that the first speed is run through to get to the reverse. This lengthens the gear box somewhat, and also has the disadvantage that one always has to go through the low speed to get the reverse.

There are several ways of having a reverse without running through the low speed. One is to have a separate lever to work the reverse, but this is not very good, as it is inconvenient to work and expensive. The most common plan is to have a third sliding part which puts in the reverse, and a separate pick up of some kind.

If the car is chain-driven the Mors plan, shown in fig. 232, can be used, and advantage taken of the direct drive without having to go through two gears on the lower speeds; but then the sliding part must be on the driving shaft, as otherwise we cannot get our direct drive into gear. As shown, this type has a run-through gear, but it can also have a gate change, as shown in fig. 233. In this latter case it can have the low speed and reverse sliding on

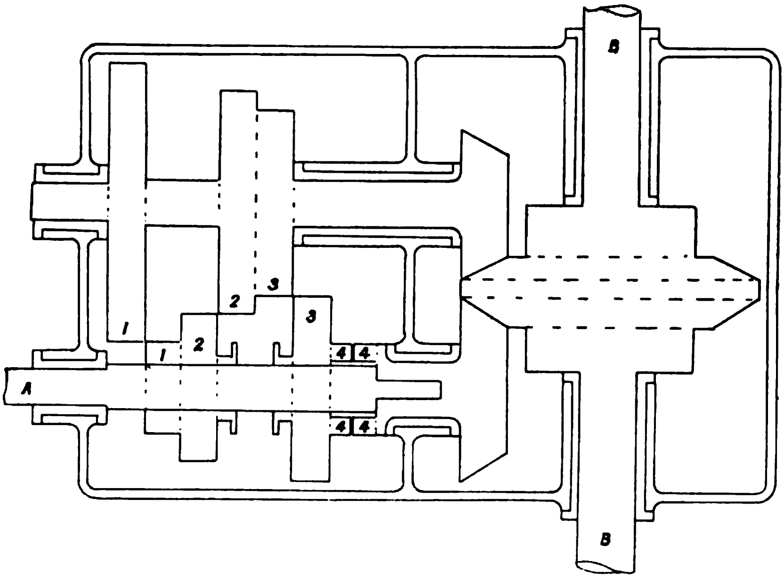


Fig. 233.

the driven shaft, as shown in fig. 234, which shows it arranged for a three-speed drive, which allows the reverse pinion being kept in gear. It could have the two low speeds of a four-speed gear box arranged in the same way. The gear can be arranged for a run-through gear, but with the bevels back to back, as in fig. 233, if desired.

This type of gear seems on the whole to have the greatest number of advantages (on a chain-driven car), but it is not quite so convenient to arrange in the matter of gear ratios as the one with a back shaft. It has, in fact, the same disadvantages as the latter when the gears are on the driving shaft. Either the whole of the gear down must be in the sliding wheels, or the back shaft must run faster than the engine on the top gear. On the other hand, as there is not quite so much loss in friction on the low gears, they need not be so low in proportion to the top as in the ordinary case to get the same tractive force.

In the matter of the arrangement of the reverse the simplest way, when the sliding part is on the driven shaft, is to have an idle wheel always running,

this is shown in fig. 237. In this case it is quite easy to arrange that it should run a great deal slower than the engine, so that there is no objection to its being always running. If the sliding parts are on the driving shaft, to keep a wheel always in gear would mean that it had either to be very large or else run very fast, so it must be slid into gear as required. A broad idle pinion may be mounted on an eccentric pin turned so as to gear with both the low-speed gears. A broad pinion may also be slid across the two low-speed gears, as in fig. 236. If four speeds and a gate-change are used it is usual to have one of the latter plans and a separate motion to work it. This will be referred to later.

As with crank cases either the top half or the bottom half of the gear box may be carried on the frame, the other one being loose—that is, either the

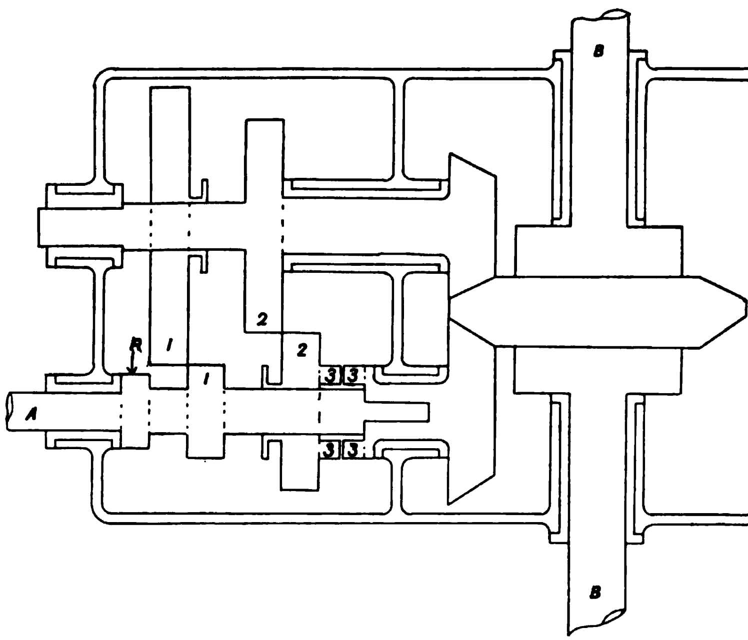


Fig. 234.

top or the bottom half may be removable for getting at the gear; the top half is generally the removable one. If the car has a live axle this will certainly be the most convenient, as a rule, as the gear box is just under the floor board of the front seat and is easily accessible. In chain-driven cars, however, the gear box is usually put much further back, so as to get a short chain, and then it is not very accessible through the floor of the car; it is then best to make the bottom half removable.

Another reason which may make it desirable to have the top part the fixed one is that it may be desired to make the casting of the crank case and gear box in one piece. This makes a very cheap arrangement to erect, as the gear box and engine bearings are all machined in line, and there is, therefore, no lining up on the frame. It has been seen that it is much more convenient to make the top half of the crank case the fixed one, so that, if

we are to make the crank case and gear box in one piece, it will probably be better to make that in the same way. The difficulty in making a satisfactory arrangement is that the flywheel has to be enclosed in the casting, and this means either a very large casting or a small flywheel. Probably the ordinary arrangement of having the two parts separate is the best for large cars of considerable power, but it seems quite likely that the plan of casting the two in one will be preferred for small cheap cars.

It is, of course, quite possible to make the bottom half of the crank case and gear box the fixed one when making them in one casting. It has the advantage of casing-in the lower half of the flywheel without any special arrangement; but, on the whole, the other plan seems preferable.

It is now usual to arrange the shafts of the gear box side by side. In many of the early cars they were placed one above the other. This was done to get a low engine with a high countershaft in the cars of that time, which were almost always shaft-to-shaft drive. There is no great objection to having them one above the other; but, if the gear box is split on the line of the shafts, it is not very convenient to make the split vertical, and, therefore, with the shafts over each other, the box has generally to be split in two places. This is expensive and heavy, as there are two flange joints.

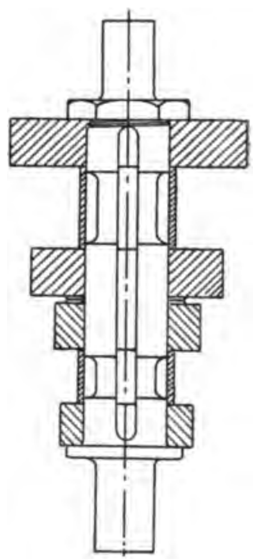


Fig. 235.

There are three ways of making the gear box itself. The usual way is to have it in two halves split along the line of the shafts. In this case there is usually an inspection cover, as in fig. 236, to allow of the gears being seen. Another way is to carry all the bearings in one-half of the gear box, the other half being an oil cover, as in the case of the crank case. The third way is to make the gear box a connected whole, with a large door in the top for the insertion of the gears. This seems the best plan for a live-axle car, as all the parts are easily accessible, and the joint in the middle of the box is avoided.

For chain-driven cars, the best plan is, probably, to have all the bearings carried on one-half of the box, and to make an oil cover serve for the other half. If the gear box is under the body of the car, the oil cover will be best for the bottom half, as in the case of the engine.

There are several ways of fastening the gear wheels on the various shafts, a common one being to make them solid with it, which precludes their becoming loose, but it is expensive, as either much material has to be machined off, or they must be made from a rather expensive forging. Besides this, it has the very great disadvantage that if it is desired to renew one gear wheel, the whole lot have to be renewed. This is very unsatisfactory, and, as far as possible, each gear wheel should be individually renewable. A common plan is to forge a flange on the shaft, and have the gear wheel simply a ring bolted on to it. This is a very good plan indeed, as far as renewals go, as the only part that need be renewed is the ring itself. It is rather expensive, however, as the flanges are expensive things to forge and machine, and there are a good many bolts to fit. There is also

always a slight liability for the bolts to work loose with disastrous results, though this should not occur with careful locking. It is also often difficult to arrange nicely, as where several wheels have to be bolted to the same shaft, the hole in each ring has to be larger than the diameter of the flange it has to go over. Probably the simplest and cheapest is to thread the wheels on to the shaft, as in fig. 235, and have distance pieces between them. The wheels have a keyway cut in them, and the shaft has a key in it. This makes the wheels very cheap to make, as they are flat discs with a hole and a keyway, and the shaft is a plain round shaft also with a keyway. The

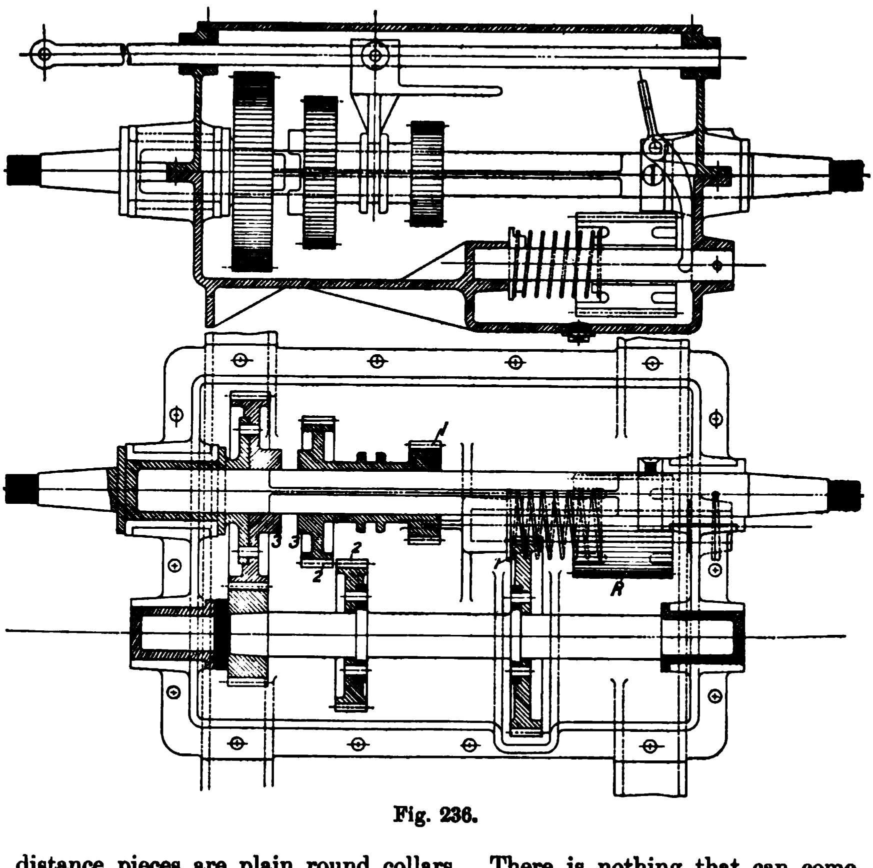


Fig. 236.

distance pieces are plain round collars. There is nothing that can come loose, even if carelessly put together by a repairer. It is usual for the sliding parts to slide on a square shaft. It is, however, not uncommon to make the sliders with keyways, and have corresponding keys on the shaft machined out of the solid; or it is possible to fit keys in the shaft, which would probably be cheaper.

With regard to the other parts in the gear box and their general arrangement, there is a good deal of margin for difference in cost and the easy renewal of wearing parts. Fig. 236 shows a gear box of fairly conventional

design, which illustrates several of the points. In this it will be seen that the sliding parts are on the driving shaft, and, therefore, the countershaft has to run twice as fast as the engine on the top speed. As a result of this, also, the reverse pinion cannot be kept running, but is a sliding pinion pushed into gear with the two low-speed gear wheels by a lever, and pushed out of gear by a spring. This makes a very long gear box, and is not very convenient to get into gear. In order to keep the gear box at all a reasonable length, the lever that pushes the reverse pinion into gear is very short at the driving end and long at the other, which causes a good deal of resistance

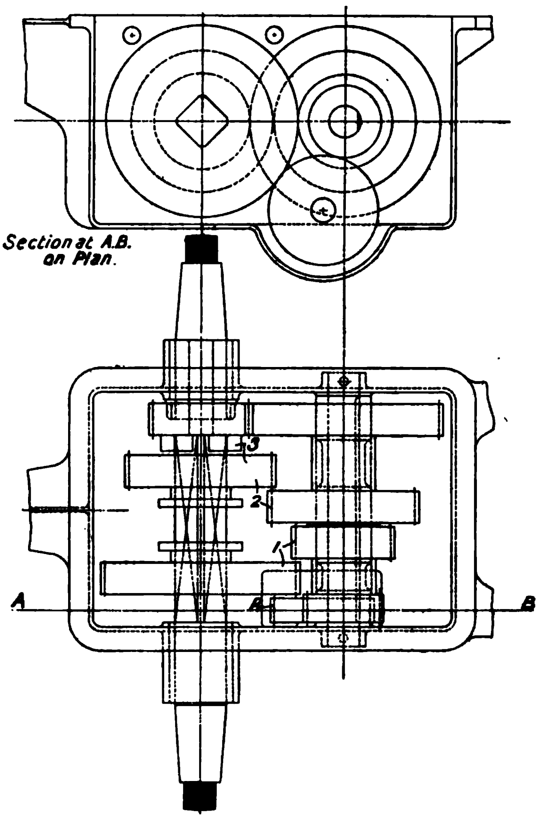


Fig. 237.

in getting the gear in. Further, the spring does not always push the pinion out of gear, and in this case one may get the low speed and reverse into gear at the same time. Both the low-speed pinions are solid on the sliding part, and, therefore, if *either* of them requires renewal *both* have to be renewed. The construction of the back shaft shows very clearly the difficulties of fixing loose rings on by flanges. In order to get over the difficulty of having to pass each ring over the flange to which the previous one is fastened, the small wheel at the end of the casting is fixed on with a cone

and nut. The flanged shaft at the driven end is a somewhat expensive forging, and as the after wheel is fastened on to it with bolts and nuts, this takes a certain amount of fitting. The great length of the shafts in the gear box also makes them liable to spring.

Fig. 237 shows a gear box with the same width of teeth, but differently disposed. In the first place, the length is reduced to a good deal less than half. This, of course, means a much cheaper and lighter casting. In order to do this the parts are differently arranged, and the sliding gears are in two pieces with a gate change. The sliding parts are on the driven shaft, and the countershaft is consequently driven at half the speed of the engine. The reverse pinion is always kept in gear, and, as it runs a great deal slower than the engine, there is no real objection to this. This makes a more compact arrangement, and makes the reverse much easier to put in gear and quite certain of coming out. It also saves the expense of the small lever and spring to work the reverse.

The gear box is not split along the line of the shafts, and thereby a joint and several bolts and nuts are avoided. The square shaft is put in from the after end and threaded through the sliding parts. The countershaft is hollow and is not fixed to the shaft, but runs loose on it. The shaft is a bronze pin. By taking out this pin the whole of the countershaft can be withdrawn through the top of the gear box, and the cover of this is large enough to properly get at all the gear. Similarly, by taking off the universal joint, the square shaft can be drawn out of the gear box, and the rest of the gears taken out.

Each wheel is quite separate, and can be renewed independently of the others. The wheels on the countershaft are plain discs threaded on to the shaft and secured by a key, as in fig. 235.

It will be noticed that the reverse in this gear box is narrower than the other speeds, and this seems to be justified, in view of the fact that it is not much used. It will also be seen that the gear permanently connecting the back shaft with the propeller shaft in fig. 236 is wider than the gears that are changed, while the gear that connects the engine shaft with the countershaft in fig. 237 is not. The latter is right, as the wear on the gears that are changed is always greater than on those that are not.

The gear ratio in both these boxes is the same, being 1 : 2 : 4.

The gear box must have lugs cast on it to carry it on the frame. The older plan of construction is to have an inside frame to carry both the engine and gear box, as in fig. 238. This is not very satisfactory, as it makes a very expensive frame. It certainly makes the parts convenient to erect, but there should be no difficulty in making them quite convenient to erect without one. Further, the gear-box in a chain-driven car is now placed so far aft that the inside frame would have to run the whole length of the car. The simplest and cheapest plan seems to be to cast arms on the gear box wide enough to extend across the main frame, as in figs. 6 and 215. This will probably also be the lightest, as the arms will be made of aluminium, and probably about as light as the steel cross-pieces that are used to carry an inside frame. It also has the advantage that the pedals can be carried on these arms, so that the gear box with its pedals, foot-brake, &c., can all be erected on the bench ready to drop into the car. Instead of doing this, there may be two cross-pieces on the frame to carry the gear box. This should be, theoretically, a little lighter, but it is doubtful whether it would really be so, while it would be more expensive, as there are more joints.

If the car is chain-driven, the general arrangement of the change-speed gear box itself is practically the same as in the case of the gear-driven car. In fact, it is sometimes made exactly the same, and the bevel gear for the cross shaft put in a separate casing altogether. In this case the cross shaft and casing are arranged exactly like the back axle of a live-axle car, and, in fact, have sometimes been a stock live axle with chain wheels at the end of it. This arrangement is shown in fig. 5. The advantage claimed for it is that the gear box can be arranged to come close to the engine, as in a live-axle car, and yet the cross shaft can be placed at the after end of the car to get short chains. Then, to alter the length of the chassis, all that has to be altered is the length of the intermediate shaft. This is no doubt true, but it is considerably more expensive, as there are four parts to the casings instead of two. It will be heavier for the same reason. There are also two

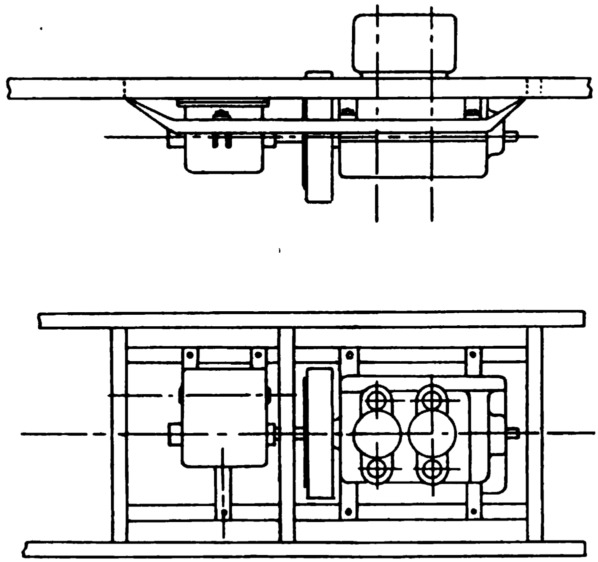


Fig. 238.

casings to erect on the frame instead of one, and this means more lugs and fastenings. If the same firm is making both live-axle and chain-driven cars, it enables the same gear box to be used for each, but in all probability makers will, in the future, specialise on either one or the other plan. Where they are separate the gear box will be carried as above, and the bevel gear case will go right across the car and be carried on the side members of the frame.

The more usual plan is to enclose the bevel gear in the same casing as the gear box, which is extended at the after end for this to be done. In this case the arrangements for carrying it are slightly different. In the older chain-driven cars the chain wheels themselves were carried on short lengths of shaft, each carried on a bearing fixed to the side member of the frame. This is shown in fig. 239. This may put a considerable twisting strain on the side members of the frame, and is expensive, as separate shafts

are required for each chain wheel, with a claw coupling to couple them to the shafts from the gear box. Then these have to be fitted up to the frame and have lugs on them to fix them by. The gear box also has to be carried on lugs of its own, and all these parts have to be lined up with each other.

A very simple plan, used with considerable success, is to carry the weight of the gear box on the cross shafts, as in fig. 216. In this case it will not want much lining up, as the after end of the box is bound to be in line with the shafts. Perhaps the best way of all is to make the gear box also a casing round the cross shafts, and make this casing carry the after end of it, as in fig. 240. This will be very little, if any, heavier than carrying out

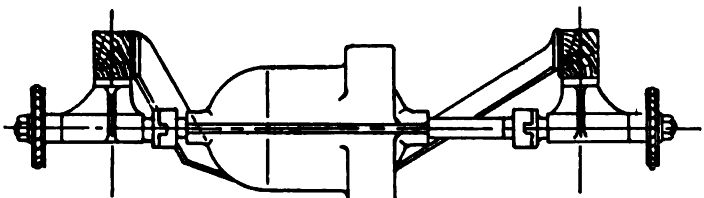


Fig. 239.

arms to carry it, and makes a very substantial job. The only slight difficulty there is, is that there is no very convenient place for fitting the brake drum outside the gear box. There is no great difficulty in making a brake to run in oil, however, if necessary, or the brake can be put at the front end of the countershaft if the drive is of the Mors type, as shown in fig. 232.

As to the general distribution of the gear in the gear box, the construction of the differential is dealt with later. The cage that carries it is usually prolonged on one side to carry a brake drum, the general distribution of parts being as in fig. 241. This shows a change-speed gear with three speeds and reverse with direct drive on top speed, exactly as in fig. 237, but

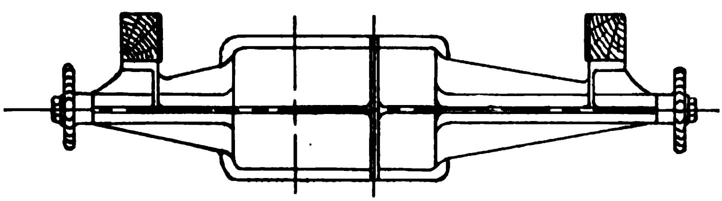


Fig. 240.

arranged for chain drive. It is not easy to arrange that the gear case can be put together, without splitting, on the centre of the shafts; but it should always be so arranged that the loose half is only an oil cover and does not hold the bearings in position.

The sliding parts are usually worked by a fork secured on a sliding rod, as in fig. 236, and this is very conveniently made of a gunmetal or malleable iron casting. Theoretically, the former is the best, as the gunmetal is a suitable wearing material to run against the steel. It should be secured on the sliding rod in such a way that there is no chance of its coming loose, as the strain under the shock of the gear engaging is somewhat greater than one would imagine. A set screw with a lock nut is sometimes used, but is

not a very good fastening, as there is a slight possibility of the set screw coming out. In this case it not only makes it impossible to change gear, but the set screw may get between the gear wheels and do serious damage. Taper pins split at the end are another fixing which is quite satisfactory, if well done. A bolt put right through with a castle nut secured with a split pin is also a safe one.

In some cases the sliding parts are moved by a swinging arm, as in fig. 242. This often lends itself to a rather cheaper arrangement of the change-speed gear. Whether it will do so or not depends very much on the general arrangement of the car in other respects. It is also claimed for it that there is no liability of getting grit into the gear case, and that the sliding

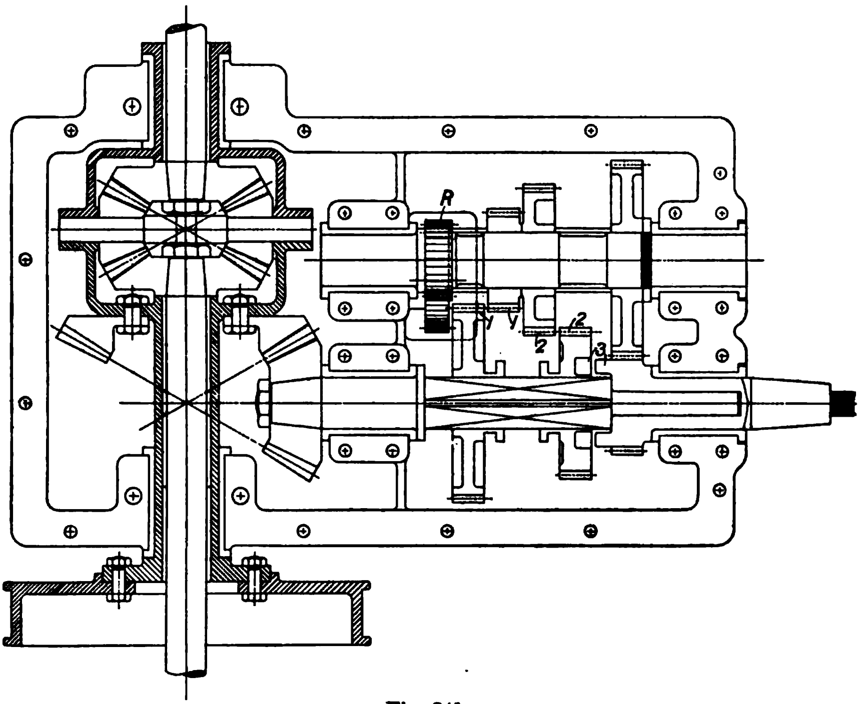


Fig. 241.

rods otherwise used are liable to carry some in every time they are moved. Probably there is little in this, and by far the greater number of cars have sliding rods. Where this form of change is used there are steel rollers instead of the fork to work the gears. In actual construction of the gear box there cannot be much difference in cost between this and the sliding rod, but where the change-speed lever is carried on the steering column it generally makes a very cheap arrangement to erect.

The arrangement of the change-speed levers varies a great deal in different cars. The present tendency is to fit them all at the side of the car, but in many cases this is certainly more a matter of fashion than anything else. Where the gear is a run-through gear they may very conveniently be carried

on the steering column. Fig. 243 shows one arrangement in diagram which is very convenient to erect, as the change-speed lever and quadrant are all erected with the steering column on the bench, and simply have to be fixed in place on the car. Then one link connecting the arm from the change-speed lever to the gear box finishes it.

This arrangement of change-speed gear has many conveniences, as well as

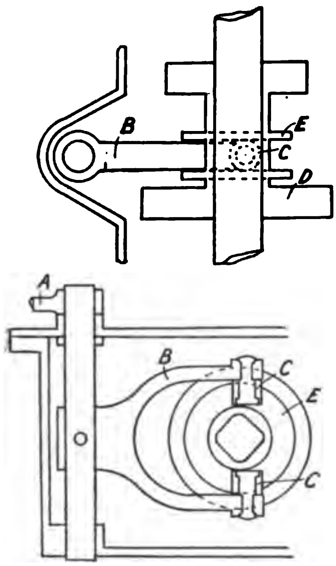


Fig. 242.

A, Arm connected to change-speed gear.
B, Arm carrying rollers, C, which move sliding part, D, of change speed by flanges, E.

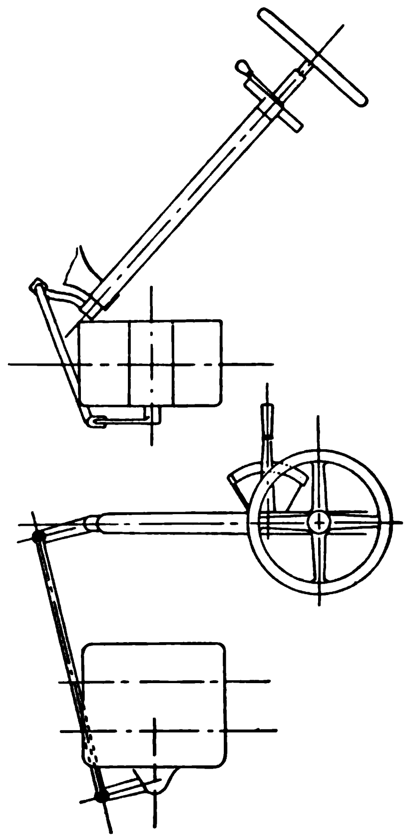


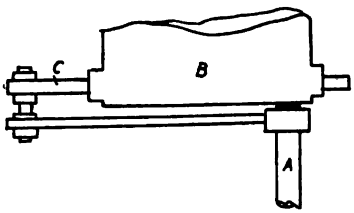
Fig. 243.

being very cheap to make. In small cars it gets over the difficulty of the change-speed lever being in the way when getting in and out of the driving side. For larger cars, however, the side lever is almost universal.

Where the gear is run through and we have a side lever, this should be arranged to work the gear with as few joints as possible. If we can get the lever behind the gear box we can make it lead straight on to the sliding rod, as in fig. 215. Where the gear box is at the back of the car this arrangement is the same reversed. If the gear box is practically alongside the change-speed lever, as in some arrangements is practically unavoidable, perhaps the best way is to arrange it as in fig. 244. In any case, an arrangement with

many joints should be avoided, as this involves extra expense, and also the joints work slack, and then the gear does not stay in the right place.

If the gear is in two halves there are several different arrangements for working it. It is, of course, necessary that the arrangement should be such that it is impossible to put two of the gears in at once. In some cars there have been two levers, one to each part, with a locking arrangement to stop either of them being moved unless the other is out of gear; this is obsolete for pleasure cars.



A, Spindle of change-speed lever.
B, Gear box.
C, Sliding in rod in gear box.

Fig. 244.

Another plan is to have the two parts moved by cams. This is not so much used as it used to be. Fig. 245 shows this, the cams being in a straight sliding part which works the sliding parts with bell cranks. The cams may also be cut on a circular rotating plate or on the surface of a cylinder.

This has the disadvantage that there are a good many parts to get loose, and, therefore, a liability for the gear to get a little out of place when they

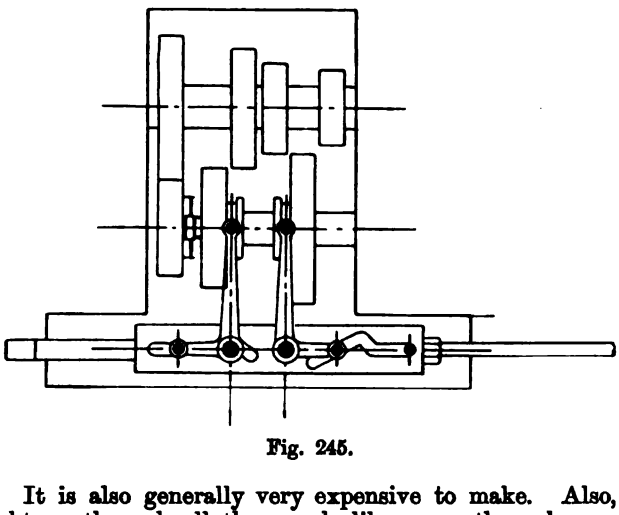


Fig. 245.

are worn. It is also generally very expensive to make. Also, the lever goes straight on through all the speeds like a run-through gear. This is considered by some people an advantage, but it does not seem to be a real one. It has now, to a large extent, gone out of use in favour of the gate quadrant.

The principle of this in its older form is shown in fig. 246. In this the change-speed lever may be moved sideways, so that it can be moved fore and

aft in either of two slots in the change-speed quadrant. As shown, the gear box has four speeds and a reverse, and therefore there are five positions necessary.

Each of the sliding rods that moves a gear has cut on it a rack, and the spindle of the change-speed lever has on it a quadrant cut to fit this. When the spindle carrying the quadrant is moved sideways so as to move in either of the slots, it will gear into one or other of the racks on the sliders. In the position shown it gears into the rack moving the first and second speed. If moved to the left it will gear into that moving the third and fourth, and if to the right into that moving the reverse. The quadrant is so arranged that the lever can only be moved sideways when all gears are out of gear, and, therefore, two gears cannot be put in at once.

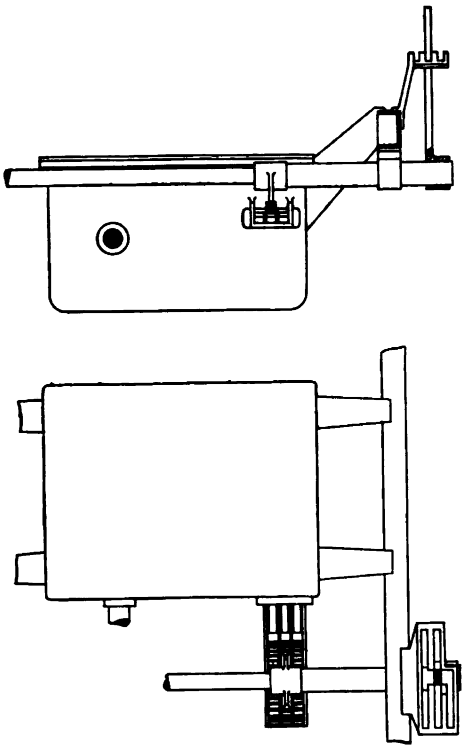


Fig. 246.

In order to keep the sliding part that is not in use still when the other is in gear there must be some lock. As there is no strain upon it, there is no reason for it to move, and, therefore, a spring lock is probably quite sufficient. If there were no lock at all it might, however, move slightly with the vibration of the car, and then when it was desired to change, the teeth of the quadrant would not gear with the rack, and it would be impossible to change gear. A very simple form of lock is a spring fitting into a notch in

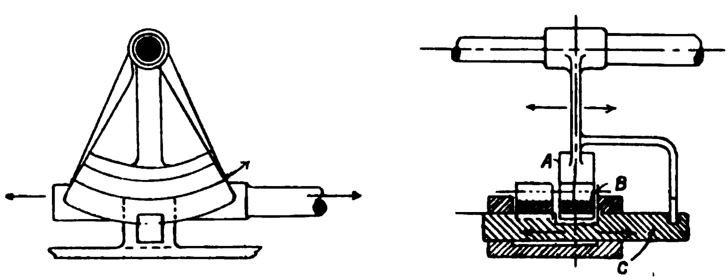


Fig. 247.

A, Quadrant. B, Rack. C, Locking piece.

the sliding rod. This is not absolutely positive, but should be quite sufficient. If there are only three speeds, a more positive lock can be made

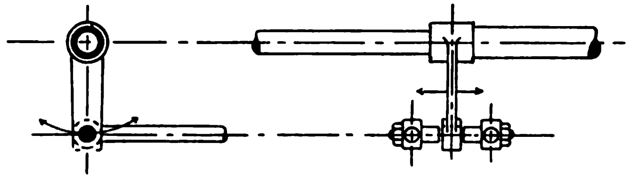


Fig. 248.

with spring catches which slide between the teeth, and are pushed out by the quadrant.

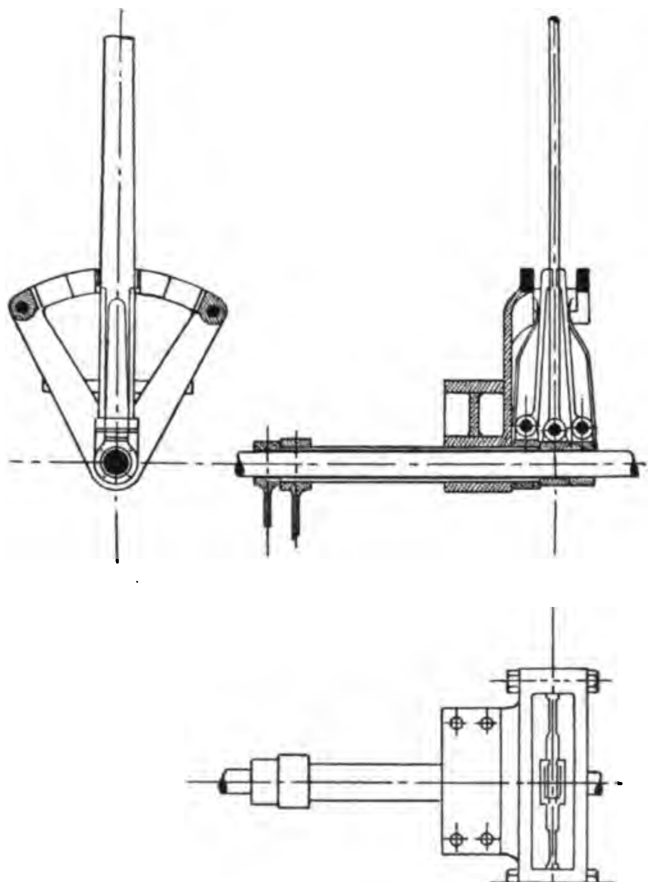


Fig. 249.

If there are four speeds a separate sliding part is generally used for the reverse, and so we have to provide for moving and locking three racks

instead of two. In this case a locking-piece may be moved along with the quadrant, something as in fig. 247, so that it locks positively all the sliding parts except the one in use. This may be varied a good deal in arrangement to suit the circumstances of the case.

Racks and Quadrants.—The racks and quadrants are expensive things to make, but may be cheapened in many ways. The most obvious is to turn rings on the sliders instead of cutting teeth. This will make it possible to finish the slider in one operation instead of about four.

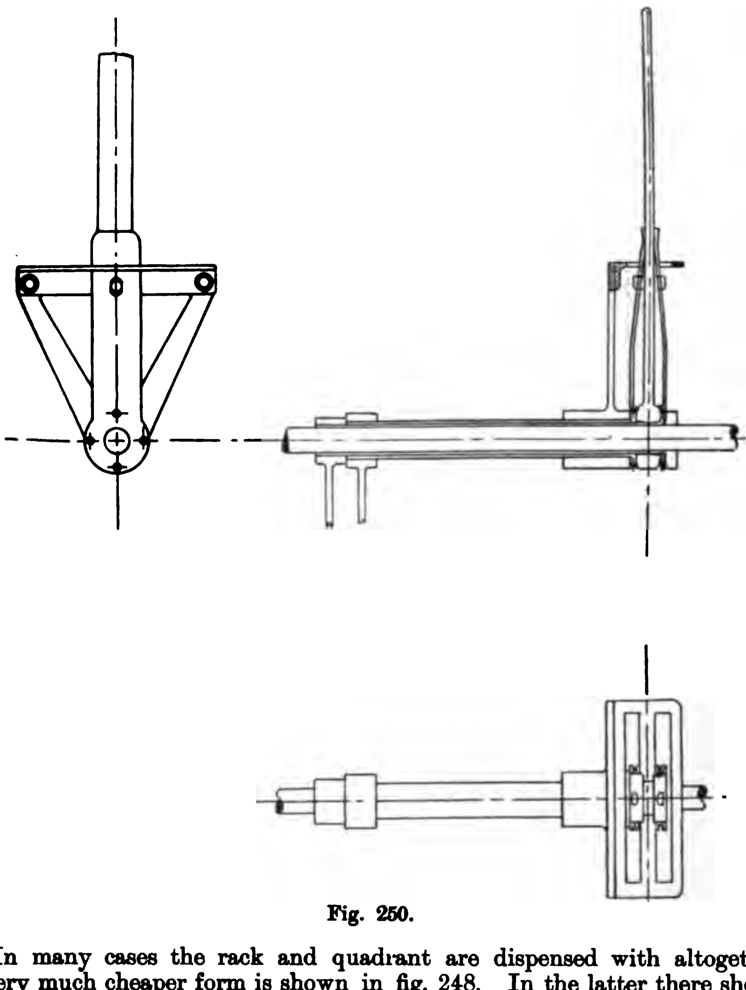


Fig. 250.

In many cases the rack and quadrant are dispensed with altogether. A very much cheaper form is shown in fig. 248. In the latter there should be no difficulty in stamping the lever and also the sliding rods so that they required no machining to make them fit, which would save a good deal of expense.

A way of arranging the cross-over gear without having to slide the shaft is to arrange the whole of the gear at the end by the lever, and to

have the two parts worked by sleeves. One arrangement of this is shown in fig. 249. In this there are two arms, each working one of the sliders while the lever itself picks up spring arms at the end of the shaft and sleeve respectively, which they are fastened on to. As shown, this is rather expensive to make, as there are three hinge joints to the lever and arms respectively, and, in addition, springs to hold the arms up to their place. On the other hand, the springs hold the lever in place in all positions, and no locking gear is required. The gear also works very free, and the slight resistance to crossing over that the gear with a sliding shaft often has, is avoided, and, therefore, the car is pleasanter to drive. A very much cheaper form of this gear is shown in fig. 250. In this all the hinges are

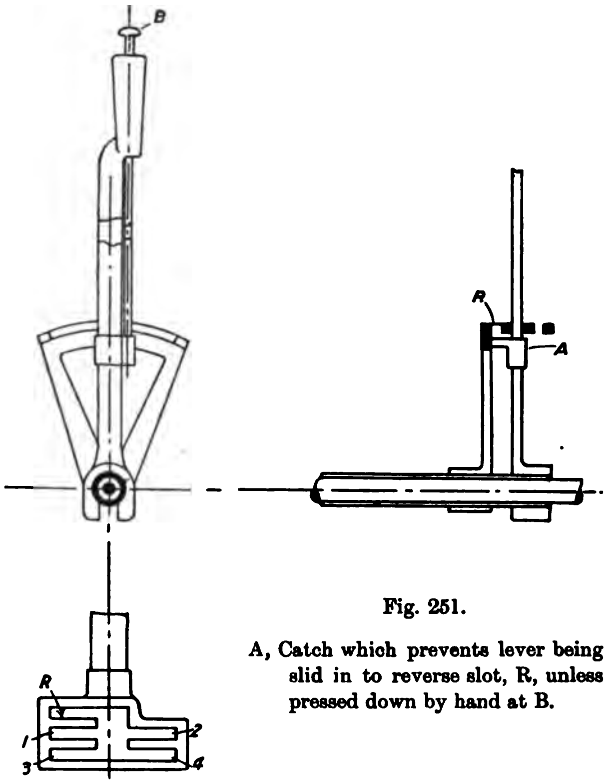


Fig. 251.

A, Catch which prevents lever being slid in to reverse slot, R, unless pressed down by hand at B.

done away with, and the springs form the arms. These are plain springs with a hole in them, and the lever has a pin which fits into this hole, and moves the arm required. The whole is carried on the spindle of the hand brake, and the two halves are worked by two sleeves, which are made of steel tube. These are opened out into a star at the end to carry the springs of the pick-up gear.

In both the last forms of pick-up gear it is not easy to arrange for four speeds and a reverse with one lever, unless the low speed is run through to get to the reverse. Occasionally a separate lever is used for the reverse, but this is expensive, and not very convenient.

Reverse Arrangements.—Where a sliding shaft is used for the change, and there are four speeds, the usual way of arranging the reverse is to have a third slot in the quadrant, as shown in fig. 251. In this case there must be some arrangement to prevent the lever being put into the reverse accidentally when crossing over. One way is shown in the figure. The button at the top of the lever has to be pressed down in order to get the lever into the reverse, while it can go into any of the four speeds without this. Another way of arranging this is to have a guard on the quadrant, which has to be lifted up, as in fig. 252.

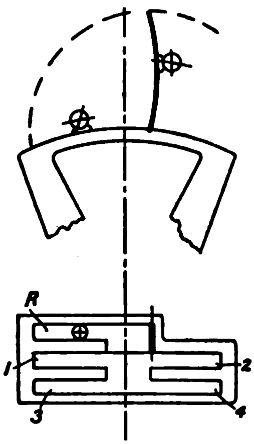


Fig. 252.

A little detail that is an advantage in the case of the cross-over gears of the types of figs. 249 and 250 is that it is easy to get the travel of the lever much more even for the different speeds than in the case of the sliding shaft. Where fairly wide gears are used the travel required for the direct speed is much less than that for the other speeds, and the reverse may also be less. With a sliding shaft the travel of the lever must be practically exactly proportionate to that of the gears. The result is very uneven travel. On the other hand, by putting the arms that work the sliders at different angles, as in fig. 253, one can make the gear (figs. 249 and 250) have practically the same travel on all the speeds and reverse.

The change-speed lever must have some means of keeping it in position in the different speeds. With run-through gear the simplest plan is to have

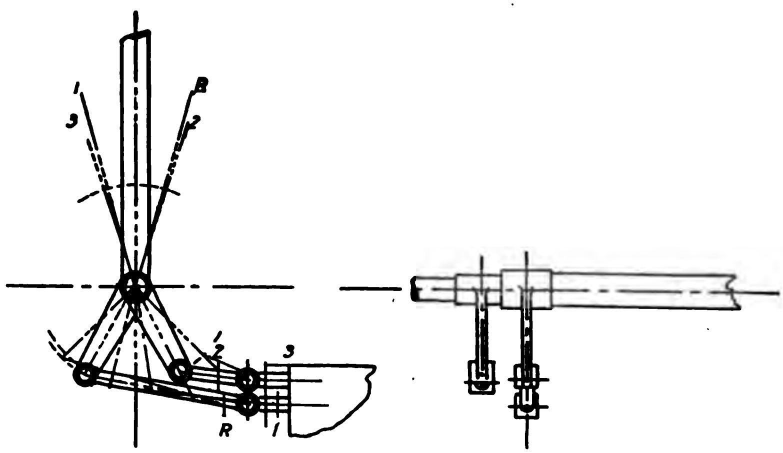


Fig. 253.

notches in the side of the quadrant, and a projection on the lever to fit into them, the lever having enough spring to come out to change gear. This is far the cheapest arrangement, and quite satisfactory. The locomotive type of reversing lever (fig. 254) is sometimes used, but has a great many parts, and

is, therefore, expensive. Fig. 255 shows a cheaper arrangement, which is quite satisfactory.

In the gate change, figs. 249 and 251 have their own locking gear. The others can generally be locked much as above.

Three v. Four Speeds.—It is evident, then, that in the arrangement of the gear box there are great advantages in the three speeds over the four. If the gear is of the run-through type the gear box has to be very much larger for the extra speed, and this entails considerably greater weight and cost (compare figs. 227 and 230). If of the gate type the extra speed, as a rule, entails an extra slider and pick-up, and this means greater expense. For all ordinary work my own opinion is that the money would be much better spent in greater engine power, which would make the car faster up hill and equally fast on the level. Besides this, the engine need not be run so hard to get the necessary power out of it. There will be less wear on an engine

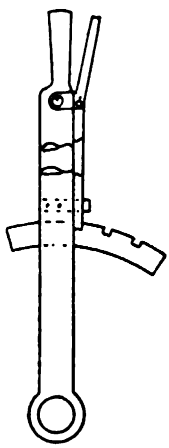


Fig. 254.

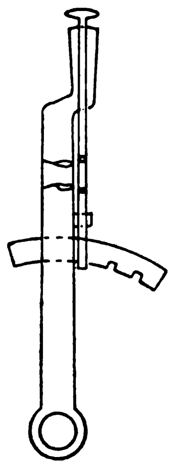


Fig. 255.

of 20 horse-power doing 10 horse-power than in a 10-horse one doing its utmost all the time.

In actual construction the gear box itself is practically always cast in aluminium. The gear wheels are generally in steel case-hardened. The hardening of the wheels is a most important point, and, in fact, entirely determines the amount of wear on them. It is evident that the strain on the teeth when the gear is changed is very great, as the clutch has to be forced to accommodate itself to the speed of the gear with a jerk. For this reason the wear on the gear depends a great deal on the weight of the moving part of the clutch, a matter dealt with under the heading of clutches. The exact hardening and tempering is, however, of the utmost importance. If the surface is too soft the gear wears very rapidly at the edges where it engages the other gear. On the other hand, if it is too hard or too deeply hardened the edges will chip or even the teeth break off. If the edge of one tooth breaks away at all, then the gear will always engage at that point and the strain will always come on the next tooth. This will soon break at the edge, and so an increasing break will occur in all the teeth till the gear is rendered useless.

The teeth must be rounded or bevelled off at the edges to allow them to go easily into gear. This, in small shops, is frequently done with a file, but in some of the larger ones it is done by a special machine. It is claimed that the latter is much superior to the former, as it is perfectly regular, and, it is said, that if done by hand it is not absolutely regular, and therefore the gears are liable to engage always in the same place causing great wear there. I cannot say that this has been my experience, and the bevelling by hand seems to me to work quite satisfactorily if well done, though no doubt where large numbers are made it is cheaper to do it by machine. The irregularity that there is likely to be in the hand-bevelled tooth is far less than the irregular wear that frequently takes place in the first few days' work of gears even by the best and largest makers.

In some cases gears are not hardened, but are made of special tough steels left in their natural state. This is a theoretical advantage, as the case-hardening of the wheels is bound to make some difference to the teeth, however small it may be, and it should be an advantage to use the gears as they are cut by the machine without heating them afterwards, and the hardness is quite uniform, as the possibility of irregularity in hardening is avoided. As a matter of fact, some of the best running gears I have ever seen were not case-hardened and they seemed to stand at least as well as those that were.

In any case, gears are generally made of special steels either specially prepared for case-hardening or to run without hardening. If they are case-hardened the running depends entirely on the uniformity with which this is done, and unless cars are made in large quantities it probably pays to get this done by those who make a speciality of it. In France, one firm makes a speciality of making and hardening wheels for the trade, and, consequently, does it very cheaply and very well.

In the actual proportions of gear wheels different makers' practice varies so enormously that it is very difficult to give any rules. Generally speaking, much wider gears are now put in than formerly, and the general adoption of the gate plan of changing speed has enabled makers to do this without increasing the size of their gear boxes. If a powerful engine, a direct drive and three speeds only are used, the top speed being sufficiently low for all ordinary hills to be taken on it, the wear on the gear is necessarily much smaller than when more speeds are used and the drive is more often through the gear. Further, the oftener the gear is changed the more is the wear on it. Thus, not only have the gears been increased in size but they are less used. If speed is the main object, gears are often made exceedingly small for the power they transmit and yet are sufficiently efficient and durable. Putting in wide gear, therefore, is not a matter of safety, but a matter of wear. Very narrow gears wear more than the wider ones, and when worn are very noisy. It would appear that with the gate plan of changing the gears might be much wider than is customary, as the increase in weight and cost is very slight. To show the variation there is in practice in this matter, it may be mentioned that 80 horse-power cars have been built with gears $\frac{5}{8}$ inch wide and 10 horse-power cars with gears 1 inch wide. The latter is probably the narrowest that should be used for a car of, say, 15 horse-power or upwards, while for the larger sizes the width should be $1\frac{1}{4}$ inches or $1\frac{1}{2}$ inches. The wider wheels give more margin of wear and run more quietly, particularly after some use.

The distance the centres of the shafts in the gear box are placed apart

varies nearly as much as the width of gear. Some makers who use very narrow gears make them large in diameter to compensate for this. Roughly speaking, the centres vary from four to six times the width of the gears, though some makers exceed these proportions. The proportions adopted depend on the general arrangement of the parts. If the gate change is used the gear box is generally compact, and a good arrangement is to make the gears rather small in diameter and wide; but if a run-through is combined with very wide gears the length of the box becomes impracticable; therefore, t is generally better to use gears of ample diameter. On the other hand, if the diameter is excessive the surface speed at which they run becomes very great and it is not easy to make them run quietly.

There is great variation in the size of the teeth. Some makers, who adopt gears of very large diameter, use very fine pitched teeth; on the other hand, some gears of quite small diameter have very coarse teeth. The tendency is to have coarse teeth, but it is not very marked. Very fine teeth are apt to scream with a high-pitched noise if the surface speed is very high, while coarse teeth, especially on small wheels, emit a grunting or groaning noise. Which is the most objectionable is a matter of idiosyncrasy, but, on the whole, the balance of advantage would seem to be with pretty coarse teeth. They are less liable to damage, much stronger, and less affected by the centres of the shafts not being exactly the right distance apart, which may happen from wear. It used to be the practice in the gear boxes with shaft-to-shaft drive to have wheels with different pitches of teeth, the size of the teeth increasing with the difference in the diameters of the engaging wheels. The result was that the smallest pinion had the coarsest teeth. Now it is more common for the teeth in a gear box to be all the same pitch, and this seems more convenient for the manufacturers. Practically, gears of about 6 diametrical pitch seem about right for small gear boxes and 5 or 4 for large ones. Sometimes the teeth are much finer, 8-pitch teeth being used by some makers for gear boxes with centres very wide apart.

Gear boxes are cast as thin as possible, the parts subject to strain being strengthened by webs which should connect the bosses carrying the shaft bearings in order to take the thrust from the gears. This is far greater than might be expected, and, therefore, these webs should be substantial, and, in particular, any wheels, such as a reverse pinion, which are carried on the flat portion of the bottom of the gear box should have good webs to carry each end of the shaft.

In order to run smoothly, the shafts in the gear box must be of ample strength to take the thrust without springing, and, therefore, should be somewhat larger in a long gear box than a short one, even though the twisting strain is only the same. The twisting strain can, theoretically, be calculated from the amount it will require to make the back wheels slip on the ground, but this must not be taken entirely by itself, as all calculations want checking by actual practice. In practice, a square shaft somewhat smaller than the crank shaft is usual.

The bearings of the gear box may be either ball or plain, and the latter may be either bronze, white metal, or hardened steel. The question of ball bearings will be dealt with later. Of the plain bearings, bronze is the most usual. Hardened steel bearings must run on hardened steel shafts, but have the disadvantage of all hardened steel bearings that, if anything does go wrong with them, it is impossible to do anything with them. White

metal has the advantage over bronze that it does not cut if small chips of the teeth get into it from the gear box. The latter is always liable to happen, unless special means be taken to prevent it, and, with bronze bearings, it is desirable to have grease lubrication, as in this case the grease is continuously squeezed through the bearing and prevents any dirt from the gear box getting in. The bearings are sometimes run with splash lubrication from the gear box, and may run all right, but, if dirt gets in, they cut very badly in a very short time. The best plan is to have a grease lubricator on the dash-board, with pipes to all the gear-box bearings. If white metal is used, the dirt apparently does much less harm, and there seems no reason why it should not be used with splash lubrication.

The grease lubrication has the advantage that it keeps the oil in the gear box from working out at the end of the shafts. This is a matter of importance, as oil is much the best lubricant for the gears.

Differential Gears.—When a car goes round a corner, the two driving wheels necessarily describe circles of different diameter, and, therefore, there must be some arrangement to allow them to revolve at different rates. Practically, there are only two ways of doing this. The one is to have the wheels loose on their axles, driven by ratchets, as in the case of many carrier tricycles; the other is to have a differential gear.

The former plan has been used in motor work, but only very little. In it the inside wheel is the one which does all the driving, and the outer one overruns the axle in going round a corner. In this plan it is not easy to get in a reverse, as some special means have to be taken to lock the clutches. Also, the only brake that can be used is one on the road wheel. For these reasons, it is not likely that this plan will come into use, though it is cheap.

There are two types of differential gear in general use:—

The bevel gear type, and
The face gear type.

The general construction of the former is shown in figs. 241 and 261. In it there are two bevel wheels, each connected to the shaft driving one of the road wheels, and there are two or more bevel pinions gearing into these. These pinions are carried on a cage and so arranged that, when the car goes round a corner, the wheels revolve at different speeds, but the mean of their speeds must be that of the differential cage.

In the face type of gear (fig. 256), the same effect is obtained by having two gear wheels on the shafts and pinions gearing into them, but in this case there must be double the number of pinions that there are in the bevel type. If the pinions could gear into the wheels on the opposite sides, as the bevel ones do, the same number of pinions would suffice; this might be done by having external and internal gears with the pinions between them; but the driving strain on one wheel would be greater than that on the other, which would facilitate slipping. Such a gear is never used. The only plan, then, is to have two pinions, each gearing into the other, in place of the one in the bevel type.

In comparing these types, the most important point to notice is that for a given-sized cage the gear wheels in the bevel type can be made very much larger in diameter than those in the face type. This is because the bevel wheels can be made nearly the full diameter of the cage, whereas with the face type room must be made for the pinions outside the gear wheels. In practice the bevel type may have the wheels nearly twice as large, and so

there is little more than half the load on the teeth. As the diameter of the wheels can be so much larger in the bevel type, the teeth can also be coarser and stronger, the conclusion is that for a given-sized cage the bevel type can be made very much the strongest, or for a given strength it may have the smallest cage. On the other hand, the face type has the advantage that we can be more certain of the teeth bearing their whole width, provided the pins which carry the pinions are strong enough not to spring at all, and so we may make up for want of diameter by extra width. There is a decided limit to this in practice, as the length of the pins has to be over three times the width of the teeth in gear, and if this is very great they will spring. It must be remembered that the load on the teeth of the pinions in gear is the same as that on the gear wheels; hence the width of that part of the pinions which gear with each other must be equal to the width of the gear wheels. Taking it all round, the bevel gear seems to be much the most

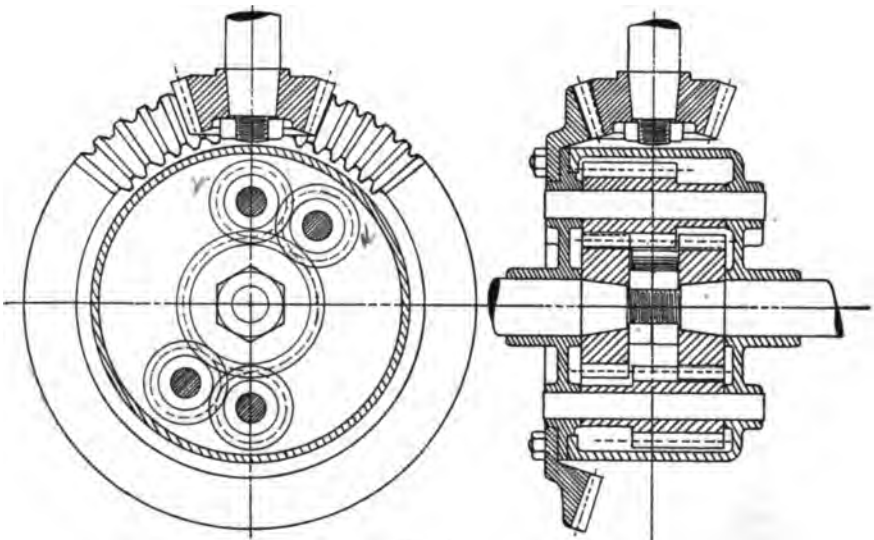


Fig. 256.

satisfactory, and probably rather cheaper. Thus, if the driving is done by two pinions, the bevel type will have two bevel wheels; two pinions; a pin to carry the pinions, which may be machined at one setting; and a cage to contain the gear with two holes for the pins, which may also be machined at one setting. The face gear will have two gear wheels; four pinions; four pins for the pinions to run on; and a cage with seatings for these four pins on four different centres. The face gears will be cheaper to cut than the bevel gears, but this is offset by the extra pins and the machining of the cage.

On the other hand, there seems no reason why the bevel wheels should be machined at all. They only revolve very slowly when the car goes round a corner; hence accuracy of fitting is not so essential for silence as it is with continuously running gear. I have used cast malleable differential gears and pinions, and they ran just as well as cut ones, but perhaps cast malleable iron is unreliable for small jobs. It should, however, be quite

possible to stamp the bevels, and in this case the material would be practically as strong as the cut gear, and for all practical purposes just as good, and cheaper even than the face gear, which would have to be cut in any case.

The cage is of malleable cast iron, and must be in two pieces held together with bolts, as shown. These are generally arranged to hold on the bevel driving wheel. The general construction is evident from the drawing.

For the customary size of differential gear wheels, see Chapter xiv. The teeth are generally as coarse as can be made without weakening the pinions too much, and are of a rather peculiar form, something like fig. 257, to give them greater strength. The wheels are all made of case-hardened steel.

It is obvious that we can increase the strength of any differential by putting in more pinions, so as to make it bear in more places. This, however, is probably not so good a plan as making the gear large enough to stand with fewer pinions, as it is not always certain that all the pinions will bear equally. Two pinions are practically necessary to get a balance, while three or four are often used. Three has the defect that none of the holes in the cage are in the same line (in the bevel type), so that both the star piece forming the pins and also the cage have to be machined on three centres. On the other hand, with two pinions they are on one centre, and with four on two.

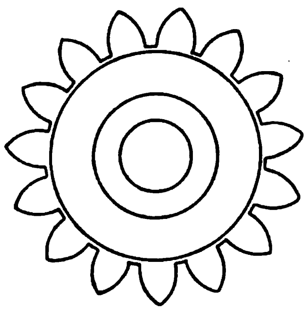


Fig. 257.

Universal Joints.—Where a live axle is employed there is a propeller shaft with either one or two universal joints to take the motion to the back axle. The shaft may either be a solid one or a piece of steel tube. The latter is not usual, but is lighter in proportion to its strength, and where it is used the parts of the universal joints can be stampings brazed into it.

Whether one or two joints are used will depend on the arrangement of the radius rods (see p. 219). If there are two joints the radius rods should be so arranged that the motion on the back one is very small; in this case a simple jaw coupling will give all the flexibility required, and is cheaper than a regular universal joint. It will also give all the end motion required.

The arrangement of the various parts should be such that the forward joint also works as nearly in a straight line as possible. It should also have as small a motion as possible, and, with the long propeller shafts at present employed, the motion is very slight.

The most ordinary form of universal joint is shown in fig. 258, and consists of two jaws with a star between them. This works very well, and is not very expensive either to make or renew. It may be varied by making the pins run in the jaws, and fixing them in the crosspiece. The wear on the pins is then slightly less, as the pressure is inversely as the distance from the centre of the shaft. It may also be arranged with the two tee pieces on the ends of the shafts, and a ring coupling them at right angles. In this case the pins can be solid on the shafts, but the ring must then be in two pieces. Some means of lubrication should be provided for this type of joint, one plan being to have a recess in the crosspiece communicating with both

the pins, and a grease cup to lubricate it. Another plan is to enclose the whole in a more or less dust-tight cover, and fill it with grease.

It is generally convenient to make the forward jaw of this type of joint in one piece with the brake drum at the end of the gear box. This type of joint works very well indeed, as the rubbing speed on the pins is very small, and yet the pressures on them can also be kept small; it does not, however, allow for any longitudinal motion. If there are two joints, this is easily provided for in the back joint; but if there is only one, some arrangement must be made for it. The usual plan is to have the after part of the joint sliding on the shaft, either on a square or a keyway. The latter is the best, as it will not slide easily on a square when there is any driving strain on it. In any case, it must be well lubricated in order that it may slide freely. A joint may be used which allows of the end motion in itself.

Fig. 259 shows the principle of one form. This may either be a four-armed star, as shown, or a two-armed one. It may also have either rollers

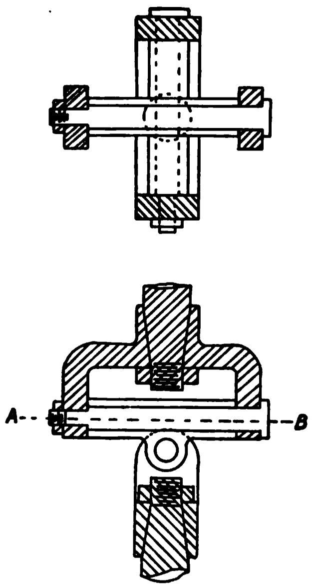


Fig. 258.

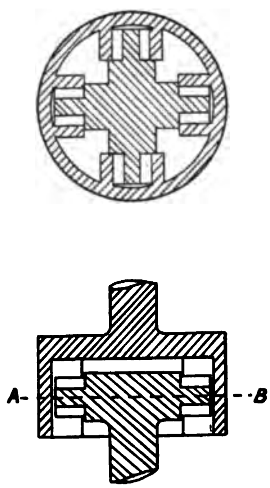


Fig. 259.

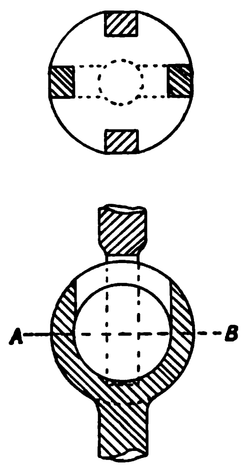


Fig. 260.

Sections taken on line, A B, in all cases.

on the arms or sliding pieces. The rollers should, in theory, give the least friction, and are certainly the cheapest to make. If sliders are used the friction must necessarily be somewhat greater than fig. 258 (with equal lubrication), as the rubbing speed is much greater; and as the pressure is lessened by increasing the distance of the sliders from the centre, so the rubbing speed is augmented. On the other hand, this type can easily be filled with grease and rendered dust-proof.

Fig. 260 shows another form of joint which is in some ways like the last, but should be cheaper to make, as there are only the two jaws and the one sphere between them, which requires two grooves turned in it at right angles. This can also be easily enclosed.

The parts of these joints are generally made of steel case-hardened. The usual way of casing them in is to wrap them up in a leather casing, and fill this with grease.

In all universal joints it is very desirable to keep the line of pressure square to the surface which transmits the power. In fig. 259, for instance, if the sliding block is very near the axis of the shaft and very wide, the pressure will be at a great angle to the surface, and consequently very great. The pressure will at all times be tangential to the radius drawn from the centre of the shaft to the point of contact, and the surface should, therefore, be kept as nearly as possible square to this.

Chains and chain wheels are almost exclusively supplied by special makers, and need not be dealt with here.

CHAPTER XIV.

AXLES—SPRINGS—RADIUS RODS—BRAKES—WHEELS.

Live Back Axles.—The ordinary type of live back axle is shown in general arrangement in fig. 261. It consists of a casing split vertically containing

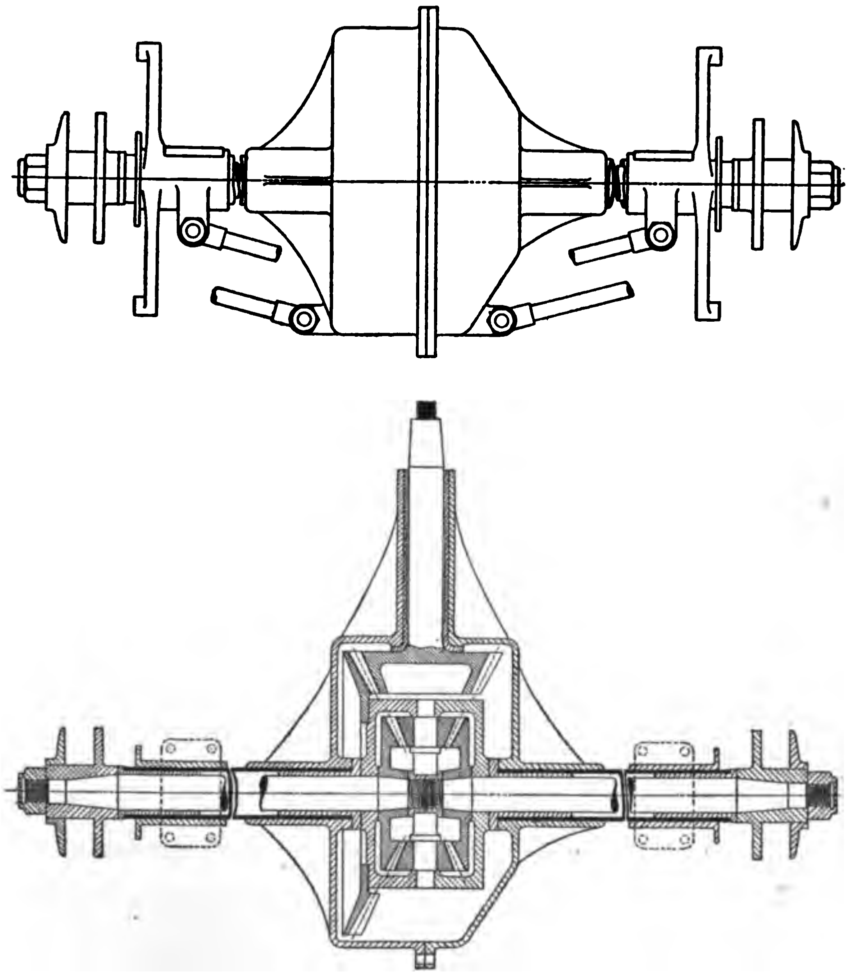


Fig. 261.

the bevel and differential gears, and carrying two tubes which case in the actual axles and carry the bearings. The actual axle is in two pieces, each

fixed to one road wheel and one bevel wheel of the differential gear. The tubes casing in these axles have malleable castings brazed on to their ends to form seatings for the springs and also to carry the brakes.

There is usually a tie-rod from these castings going under the casing, as shown, to strengthen the whole; but it is rather doubtful whether this is really needed.

Figs. 261 and 262, showing the differential gear, casing, &c., will illustrate the principal differences in detail between different axles. One difference is that in fig. 261 the bevel pinion is overhung and in fig. 262 has a bearing at each side of it. Provided that the bearing is long enough, fig. 261 appears to be the best. In the first place, it allows of the casing being made in two parts only, whereas to get fig. 262 together nicely the outside bearing has to be a third piece bolted on. This involves additional expense and weight, and another joint to leak oil. In the second place, having a bearing on the inside of the pinion involves placing the differential gear out of

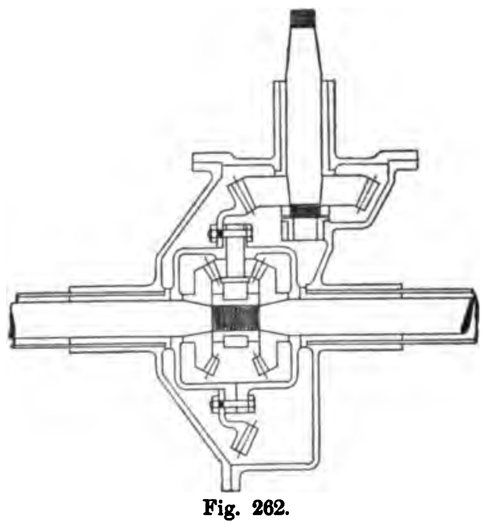


Fig. 262.

centre of the pinion shaft to clear it. This means (in a direct-drive car) that the two ends of the axle are of unequal length and, therefore, not interchangeable. In practical work the bearing on the inside of the pinion receives most of the stress, and it is difficult to make it sufficiently large.

The pinion may be either solid with the shaft, as in fig. 261, or be put on with a cone and nut, as in fig. 262. The latter is probably slightly the cheaper, and it is possible to renew the pinion and shaft separately. The former will be slightly the lighter of the two.

It will be noticed that while the two designs have casings of the same outside dimensions, the gear wheels, both of the differential and bevel gear, are wider and larger in effective diameter in fig. 261 than in fig. 262. As the whole difficulty in making cars of reasonable weight and durability is to provide sufficiently large gear wheels, &c., and at the same time, to have casings of reasonable weight, it appears that every effort should be made to get the largest and widest wheels into the smallest casings, and, therefore,

that fig. 261 is a very much better arrangement than fig. 262. Both in main drive and also in the differential gear, the stress on the teeth is necessarily high in this type of axle, so that it is important to keep it as low as possible.

If the shaft carrying the pinion is carried in a long plain bearing, as shown in fig. 261, this should be recessed in the middle to prevent its wearing to barrel shape. It would also be an advantage to recess it into the pinion, but this rather limits the range of pinion which can be used without a special bearing. If the shaft runs in a long tube forming the radius rod there is no necessity for a very long bearing here, as there is one at each end of the tube.

The differential wheels are generally fixed on to the two ends of the axle by cone and nut. It is possible to use other means, and perhaps the cheapest is simply to key them on; but there must then be some means, such as a collar on the end of the shaft, for taking the end stress, and preventing the bevel wheel from being forced off the axle, which might then come out of the car.

The casing, which contains the driving and differential gears, is generally made of cast malleable iron; but might be a trifle lighter if made of bronze. Aluminium has been tried, but it is too soft for the satisfactory fixation of the tubes. The tubes of the axle must also have castings or stampings at

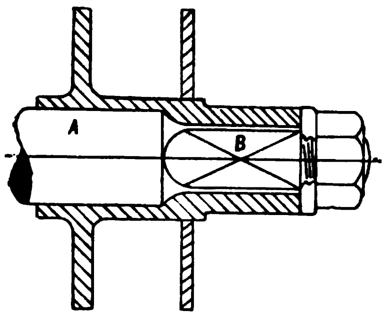


Fig. 263.

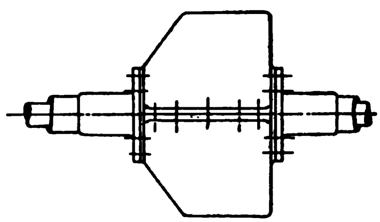


Fig. 264.

the ends to take the brakes and the springs. These should be so arranged that the two ends are interchangeable, so as to enable them to be made in larger quantities.

The wheel is generally fixed by inserting the end of the axle into the hub; the part (A) of the axle (fig. 263) has parallel sides, while the portion (B) is squared to prevent the wheel from turning on the axle. The extreme end has a screw thread cut on it for the nut, which locks the wheel in tightly.

This does not seem such a good fastening as putting it on with a cone and key, as in fig. 261, and will be more expensive.

The great objection to the above types of axle is that the axle has to be entirely removed when any inspection or repair seems to be required. As removing and replacing may cost about £2, serious defects are often neglected. In fact, it is largely the want of accessibility in the ordinary type of live axle that has given this construction a bad name.

This is partially remedied by the plan shown in fig. 264. The casing is split on the line of the axle, instead of vertically, so that one part can be taken off and the gear examined without disturbing anything else; but the drawback to it is that the tubes cannot be permanently fixed into the casing,

and have to be fixed on to it by flanges, which might be a source of weakness.

A further improvement is shown in fig. 265, by making the casing in three pieces, the top piece being an easily removed oil cover.

Another plan, which has not been much used, is to make the whole axle casing a casting split along the line of the axle. The top is arranged to take the whole of the stress, and to carry all the bearings, much as is done in the crank case of the engine; and the bottom half is formed by an oil cover. This would probably be a somewhat more expensive way of making the axle casing, and a little heavier; though not much, if made of suitable aluminium alloy. It would, however, allow of very easy inspection of the whole of the inside of the casing.

Some firms put an "inspection" door to the ordinary type of axle. As it is only a few inches square, it is impossible to wipe away the oil and dirt, which must be done before a proper inspection can be made.

A plan has been introduced by some firms of letting the wheels run on the outside of the tube, and not carrying the weight on the rotating axle, which only transmits the twisting strain. This arrangement is shown in fig. 266. It is claimed for this that it is superior, as there is a difficulty in

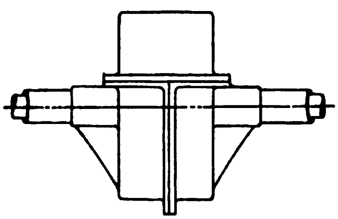


Fig. 265.

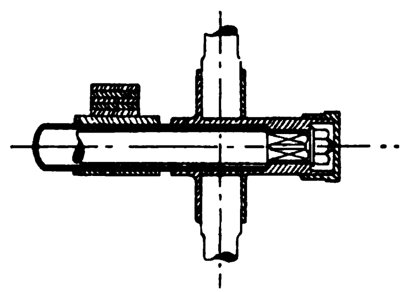


Fig. 266.

making the axle itself large enough to carry the "enormous" weight of a moderately large car. This construction does not seem to be as good as the one with the wheel fixed to the revolving axle, if the latter is made large enough for its work and of good steel. The stress is taken off the revolving axle, it is true, but it is put on to the tube, which must be thickened a good deal in consequence. The bearing on which the wheel runs must be much shorter, and, therefore, the effect of wear will be to make the wheel wobble much more than where the bearings are half the width of the car apart.

In case of accident the tube will be much more liable to damage than the more substantial axle, and will be much more difficult to repair. Further, if the wheel is fixed on a revolving axle, any damage will be very evident, but will do no serious harm, as the wheel, though wobbling in appearance, will be revolving on its axle square to the motion of the car. In the case of the wheel running on the tube, however, it will appear to run true, but will not be square to the car, and consequently will wear the tyre badly.

It might be thought that there would be a considerable saving in weight by taking the stress of the load off the revolving axle. As a matter of fact, this is not so, as, if the axle is strong enough to take the twisting strain, a very small addition to its size makes it strong enough to take the carrying

strain. For instance, a car weighing 1 ton, with 32-inch wheels, will require an axle 1·4 inches in diameter to take the maximum possible driving strain, allowing a stress of 12,000 lbs. per square inch. If the overhang of the wheel centre from the end of the bearing is $1\frac{3}{4}$ inches, the axle must be 1·53 inches in diameter to carry the weight as well, allowing the same stress. The difference in weight will be under 5 lbs., assuming the wheel gauge is 4 feet 6 inches. Considering the extra strength required in the tube, and the larger diameter of the bearings and hubs, it is evident that the construction with the wheel running on the tube will be the heavier of the two, especially where ball bearings are used.

If plain bearings are used, these should be of large size and some arrangement made for lubricating them. Ball bearings are most used, and these will be referred to later. In determining the sizes of the various parts, that of the casings and casing tubes will depend on the diameter of the axles and gear wheels. The axles are usually loaded to a maximum twisting and bending stress of about 10,000 lbs. per square inch (see Chapter xix.), and the diameters vary from about an inch in very light cars to an inch and three-quarters in large heavy ones, the average being—

Weight per axle loaded,	10 cwts.	15 cwts.	20 cwts.
Diameter of axle,	$1\frac{1}{8}$ to $1\frac{1}{4}$ inches,	$1\frac{3}{8}$ to $1\frac{1}{2}$ inches,	$1\frac{5}{8}$ to $1\frac{3}{4}$ inches.

The diameter of the crown wheel of the bevel drive varies from about 8 inches in small two-seated cars to 13 inches in large powerful ones. The tendency is to have fairly coarse teeth in the bevel wheels, the crown wheel having about 60 teeth. If a very great reduction of gear on the bevel gear is required, more will be needed in order to get a reasonable number on the small bevel; but great ratios of reduction should be avoided.

A very important point in designing live back axles, which is sometimes neglected, is to so arrange them that they can be used for a great variety of gear ratios without alteration. This is mainly a matter of so arranging the casing that the bevel wheels will not run foul of it with any of the ratios desired. Probably it will not often be necessary to use a lower gear ratio than 2 to 1 except for a purely racing car, which would, of course, be specially built in every part, and it is not convenient to have a ratio greater than 4 to 1 on account of the difficulty of making the gear run quiet.

It is well, however, to allow plenty of margin in the choice of gear, as it may easily enable one to satisfy the special requirements of a customer without making special patterns.

In designing the outside ends of the tubes which carry the axles, care should be taken that the latter do not overhang more than necessary, or the stress on them may be unnecessarily increased.

Dead Axles.—The front axle of all cars and the back axles of chain-driven cars are stationary; in such cases the wheels revolve on the axle. There are three principal ways of making an axle:—

1. It may be a solid forging.
2. It may be forged of H-section; or
3. It may be a tube.

Of these the solid forging seems slightly the most popular, while the tube is not so much used, except for the smaller cars. In considering the merits of these different methods of construction, it must be remembered that there are many stresses on an axle besides those due to the vertical load. If the

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latter only is considered, it is evident that the H-section is right, as it is, practically, the section of a flange girder. As a matter of fact, however, axles are seldom made in accordance with the theoretical stresses due to the load. If these only had to be considered, the section of the axle ought to be largest between the springs, and from there diminish to the centre of the wheel; but, as a matter of fact, the large majority of solid axles are considerably larger in section from the spring seating to the steering point than they are between the springs, and this is, no doubt, the result of practical experience. Although, possibly, this is exaggerated in some cases, there can be no doubt that there are many local stresses due to the roughness of the road, &c., outside the springs which cannot be calculated. This being so, it is evident that the theoretical advantage of the H-section need not have much weight. The H-section axles in use, which are certainly more expensive than solid axles, appear to be as heavy as the solid forged ones used in cars of the same weight, and, as the latter stand perfectly, there does not seem to be any advantage in the former. Possibly, as the result of adequate experience, the H-section type will be made lighter and yet be sufficiently strong.

The tube, on the other hand, has the advantage that it is equally strong in all directions, and in all cases is stronger for its weight than a solid bar. It ought therefore, theoretically, to be the best axle of all, and, if properly arranged, there seems no reason why it should not be. In order to be successful, however, it must be of large diameter and not too thick. Great discredit has been brought on all tube construction in cars by tubes being used which were too small in diameter for their work, and which had in consequence to be almost solid to stand.

The objection to the tube is that it cannot have parts forged on to it. Consequently, it must have ends brazed on for the wheel spindles or steering pivots, and also seatings for the springs. These should give no trouble, if well done and suitably designed. They add, however, to the weight, and it is, perhaps, doubtful whether, taking this into consideration, a tube axle can be made much lighter than a solid one, but if the parts were made of neat stampings, probably it can. In cost, where sufficient quantities are made for all parts to be stampings, the tube should be the cheapest, especially if the design is such that the tube is straight.

The steel tube has the advantage that the tube itself can be made of any quality of steel desired, as it does not have to be forged. On the other hand, with the solid forged axle the material must be such as will stand a good deal of forging, and in many cases it is found that a good quality of wrought iron is preferable to any kind of steel. The cheapest way of making a solid forged axle is to forge the ends separately, and weld them together in the middle to the exact length required. This has, however, the disadvantage that reliable welds cannot be made in many of the stronger steels, and also that the weld comes where there is the greatest stress.

Fig. 267 shows a forged back axle for a chain-driven car of ordinary design. If an H-section is employed the design is generally similar, except that the middle portion is of this section instead of being octagonal. The road wheels generally run on ball bearings in modern cars, and these will be dealt with later; but where plain bearings are used the hubs may be either of the type shown, or the wheels may be held on by flanges on the spindles and washers, as in fig. 268. There seems little advantage or disadvantage in either of these.

If the axle is a tube the spindle in which the wheel runs may be brazed into it, and the flange may be cast on the spring seating to carry the casing for the brakes. This is not a very good plan if it is desired to case-harden the wheel spindle, as it is better that this should not have to be heated for brazing. A better plan is to braze on a stamping or malleable casting forming the seating for the springs and arms for the brake, into which the wheel spindle is fastened with a cone and nut. In this case the spindle can be easily renewed in case of wear or accident.

If ball bearings are not used, the spindles are generally case-hardened and ground after hardening, and the wheels have bronze bushes. There is no doubt that hardening the spindles greatly increases the time they will wear, but it has the disadvantage that, if a spindle becomes the least bent by accident, the hardened surface is pretty certain to crack, and, if it is used after this, the crack will probably grow through the spindle in course of time, and it will break. In case of damage to such spindle great care should be taken to see that there are no cracks before repairing; if there are cracks the spindle should be condemned. Considering the time which unhardened spindles will wear, and the ease of renewal in most cases, it is doubtful if

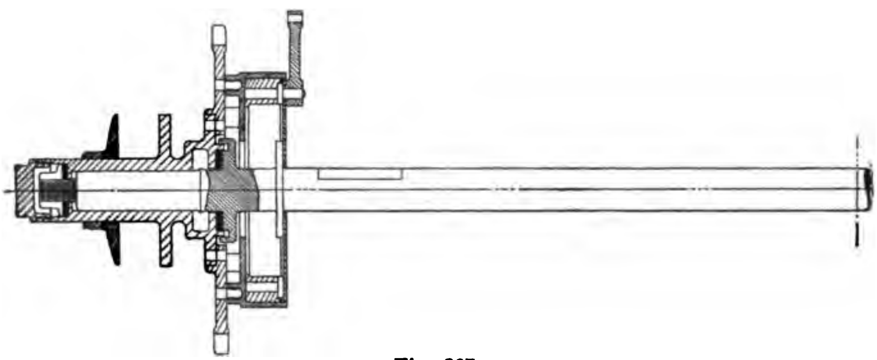


Fig. 267.

they are not preferable for this reason. In ball bearings the hardened part is a sleeve fixed to a soft spindle.

If it is desired to have a very low car the back axle can be cranked, so that the frame can drop between the wheels. As mentioned previously, it is not necessary to make a car very low, unless it is intended to drive it much faster than is desirable in this country, hence the cranked axle is not often required. It makes the axle heavier in proportion to its strength.

Front Axles.—The construction of the front axle is generally similar to the back one of a chain-driven car, with two exceptions.

1. It has pivots for the steering wheels.

2. The springs come much nearer the centre of the car, and this increases the stress on it.

In the case of the back axle, it is usual to put the springs pretty well as far out towards the wheel as possible, so as to reduce the stress on the axle. In the front axle, on the other hand, the springs must come a good way in from the wheels, in order to allow of the latter turning to a large enough angle for turning sharp corners. The tendency is to turn the wheels of modern cars to greater angles than used to be the case, so as to enable them

to turn in reasonable space with the great lengths of wheel base at present in use. This necessitates putting the springs further from the wheels, and, therefore, greater stress on the axle.

Fig. 268 shows the arrangement of an ordinary forged axle with the steering pivot running in jaws. The seatings for the springs and the jaws are forged on.

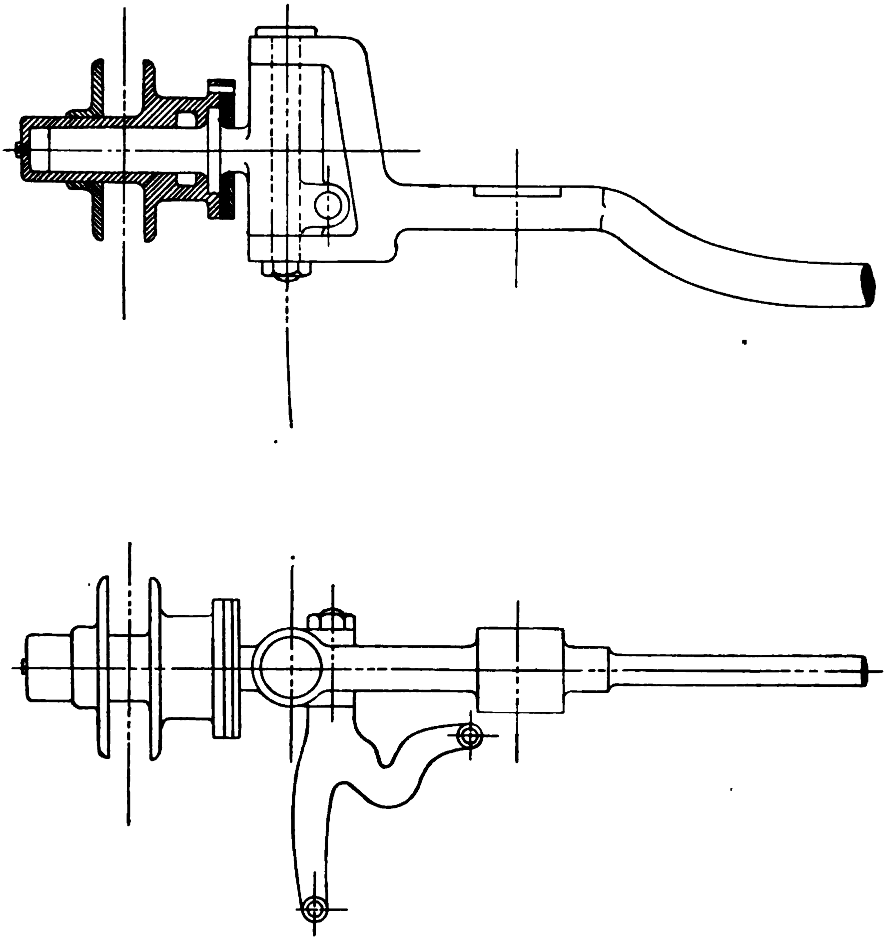


Fig. 268.

This type of axle is in general use, and is very satisfactory. If H-section is used the construction is generally similar, but the middle part is of this section, instead of being solid, and very often the jaws are also of H-section. This, in theory, makes a much lighter axle for its strength, but, as mentioned, most of the H-section axles in use do not appear much lighter than the solid ones.

The greatest variation in front axles is in the arrangement of the

steering pivots. The plan of carrying the pivot between jaws, as shown, is the most general, and is very satisfactory. It is very strong, as the pivot is held at both ends, and is, therefore, in double shear, but it makes an expensive forging of the axle. It has the advantage that the middle portion of the axle can be brought a good deal below the centre of the wheel, and this makes it easier to get in a straight axle.

The principal variations are shown in fig. 269, which is not so much

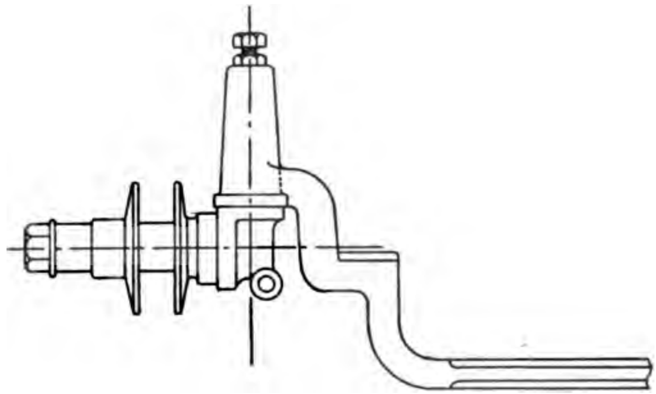


Fig. 269.

used as it was. In this the steering pivot is a long way above the centre of the wheel, and, therefore, the axle has to be carried down a long way in order to be clear of the engine. This makes an awkward forging, and is heavy. If inverted, as in fig. 270, the arrangement is much neater. In this case the end of the axle is a very much simpler forging, or may be made a very simple end for a tube axle as shown. The weight is also taken by a

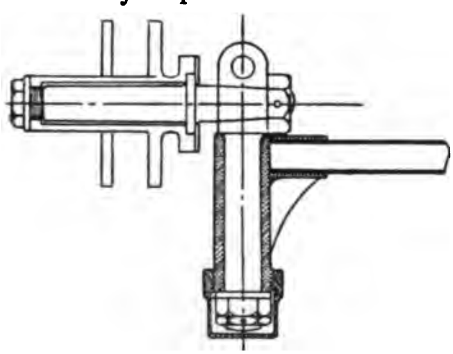


Fig. 270.

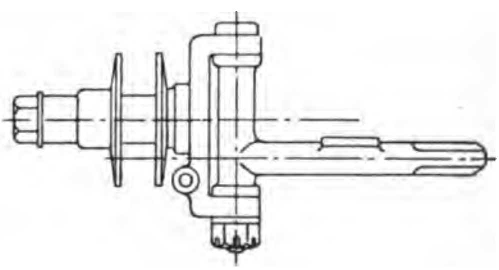


Fig. 271.

collar running in an oil-tight cap, so that it is always well lubricated. The axle can be got as far below the centre of the wheel as desired without any difficulty. A variation of the jaw form is shown in fig. 271, in which the jaw is on the wheel spindle instead of on the axle. It is possible that this is easier to manufacture, as the jaw can be very easily stamped, and the axle is a simpler forging, but there is very little in it.

If the front axle is a tube, the springs must have seats, but these are so far from the wheel that it is not easy to combine them with the end of the axle; hence special seats must be brazed on.

The wheel spindle and steering pivot can either be in one piece or separate. The former plan is usual, but the latter has the advantage that the parts can be turned out of a bar in an automatic lathe, and are very much easier to straighten if bent by accident, as each part can be set true in a lathe. Fig. 270 shows this construction.

The hubs and bearings of the front wheels are similar to those described for back wheels.

It is usual to arrange the front wheels so that they slope outwards at the top, as in fig. 272. It is not quite clear what the advantage of this is, but it is supposed to bring the centre of the steering in a line with the point at which the wheel touches the ground. This would theoretically diminish the tendency of the roughness of the road to turn the steering, and so make it steadier; but as a practical fact it does not seem to make any material difference, and if the wheels are sloped anything like enough

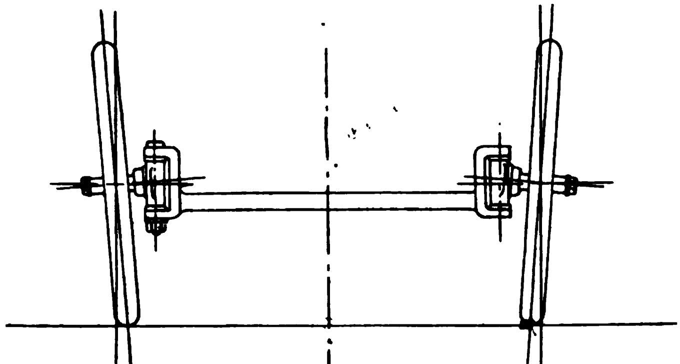


Fig. 272.

to bring the point of contact with the ground in line with the steering centre, they have a very bad appearance. In fact, to many people they look bad if sloped at all, as they are not true with the back wheels. The same effect is obtainable with the wheels vertical and the steering centres sloped to the point where they cut the ground.

Sometimes the steering centre has been placed inside the hub, but this entails the bearing of the wheel being very large in diameter and the rubbing speed very high; as also very short steering centres.

The steering arms must be fastened to the steering spindle in such a way that it is absolutely impossible for them to become loose. The most obvious way of doing this would be to make the whole of the spindle and arm in one piece, but this would make a very complicated forging. The usual way is to fix them in the way indicated by fig. 268. If the wheel spindle is separate from the steering spindle, it is perhaps cheaper to fix them in the way shown by fig. 273.

The steering arm on one side must have an arm to take the rod from the steering gear, and a second arm to take the rod connecting the two. This is generally forged on the arm that takes the connecting-rod. The arms are

best made stampings when cars are made in sufficient numbers, and are then best made of channel section for lightness and strength.

One of the most important things in arranging the steering of the car is to be sure that the steering centres have the right slope. In order to make the car steer well it is necessary that the line of the steering centre should cut the ground a little in front of the place where the wheel touches it. It does not appear to make any great difference whether this is arranged by sloping the steering centre, or by making it vertical and having the centre of the wheel a little behind it; but the former is generally the simplest arrangement. The effect of this is to make the front wheels have a tendency to run straight, and the greater the slope the greater the tendency to run straight. If the slope is too great, a great effort will be required to make the car turn a corner. If the point at which the centres of the steering spindles cut the ground is behind the place where the wheel touches it, the front wheels will always have a tendency to run crooked instead of straight. The result will be that the car will always be going to one side or the other to the extent of the backlash in the gear, and the steering will, in fact, be in unstable equilibrium.

The exact rake of steering pivots which makes the most easy steering is best found by experiment for any particular car. In some cases where the rake is too great the steering wheels have a tendency to wobble the whole time the car is running.

The slope of the steering pivots, which makes the car have a tendency to run straight when going forward, will generally have the opposite effect going backward, but this is not of any consequence.

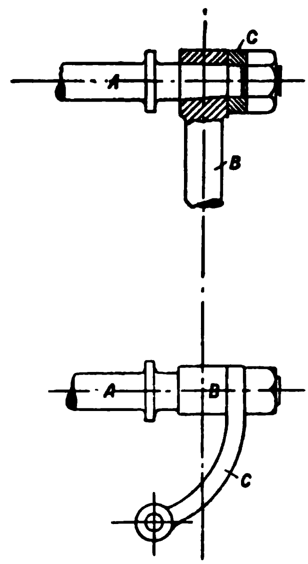


Fig. 273.
A, Wheel spindle.
B, Steering spindle.
C, Steering arm held on by same nut as secures A to B.

In arranging the front springs the effect of their compression on the slope of the steering is worth considering. If one end of the spring is very much longer than the other the slope will be varied with the motion of the spring, and this may effect the steering. In most cars the whole of the load is practically carried on the back springs so that the compression of the front springs will be fairly constant, but even then there is the question of the roughness of the road.

It is absolutely necessary that the front wheels should be set to run correctly, otherwise the wear on the tyres will be very serious. In some cases they are set to run very slightly towards each other, presumably with the idea that there may be a very slight spring in the front axle, and that this will make them run parallel. In most cases, however, it is correct to set them quite parallel. In doing this it is best not to trust to setting the wheels, as these are not always absolutely true, but to set the actual wheel spindles with a straight edge, as in fig. 274.

Springs.—In order to make a car ride comfortably over ordinary

English roads, it is absolutely necessary that it should be fitted with long easy springs, and that these should be suitably tempered. Taking the question of dimensions and arrangement first, it will be observed, on looking at any ordinary carriage, that it is carried on *double* elliptic springs having a length of 4 feet or more. In many of the smaller cars, on the other hand, the springs are only *single* elliptic and some 3 feet long. Now, a double elliptic spring has twice the range of motion that the single spring has, and it will be seen that, in this case, the range of motion of the carriage spring is several times greater than that of the car spring. There seems no reason whatever for this, as, although the carriage will have solid tyres as a rule, it has much larger wheels and does not go so fast. To ensure equal comfort, the springs in a motor car should probably be as long as those in a carriage. The delay in their adoption may be due to the influence of racing cars, as the springs preferred for these are quite different from those required for comfort at moderate speeds on rough roads. For racing, the problem is simply to get the lightest possible car for its horse-power, and, therefore, such things as springs are made as small as they possibly can be, in order to put weight into the engine. Again, the problem is quite different, since racing cars run on good roads, and are required to go round corners with the utmost speed without capsizing; this means pretty stiff springs to prevent

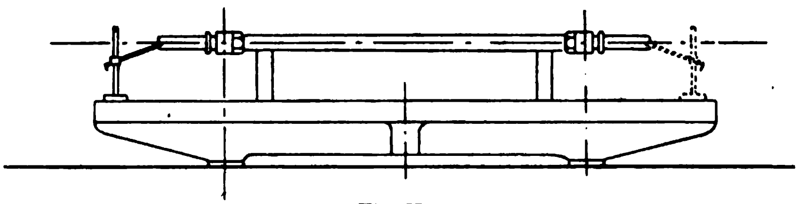


Fig. 274.

the car heeling over with the centrifugal force. The ordinary car user has no inducement for turning corners at a dangerous pace. Racing cars, therefore, have comparatively short stiff springs, and, as long as racing fashion entirely ruled the trade, these were also fitted to the touring cars. The Americans, it is true, always built their cars with long easy springs, even in their cheapest and lightest cars.

A car for easy riding on rough roads will want longer and easier springs than for fast work on very good roads. This has also been the experience in locomotives, in which springs suitable for use at high speeds on first-rate tracks in England have not proved suitable for more moderate speeds on rougher tracks in the Colonies.

In the larger cars there has been a very great improvement in the length of springs within the last year or two, but, in the smaller cars, there is often great room for improvement. This is a great pity, as the difference of cost of long springs and short ones is very small, and might be saved in other parts of the car with advantage, while in many cases it completely spoils their comfort. The Americans showed long ago that it was quite possible to make the lightest cars ride very smoothly if they were properly sprung, and also that such cars could be produced at the lowest prices.

Single springs more than 4 feet long are inconvenient, especially in the case of back springs. The front springs present less difficulty. In the first place, there is no necessity to make them as long as carriage springs in the

ordinary arrangement of car, as the passengers are almost entirely carried over the back axle. Springs from 3 to 4 feet long seem to answer all the requirements in front, and, by carrying out the frame, these can quite well be got in. At the back, however, to give really comfortable riding, they ought to be a good deal longer than this. If a longer spring is used, the frame must be extended so far beyond the body as to be both disadvantageous and unsightly. Fig. 5 shows the springing of a car with single elliptic springs, the forward ones being $3\frac{1}{4}$ feet long and the back 4 feet. It will be seen that to lengthen the springs more would be inconvenient. The simplest way of making the length more effective is to use double elliptic springs. This is shown for the back springs in fig. 215. It seems a very good plan in most ways, though it has the disadvan-

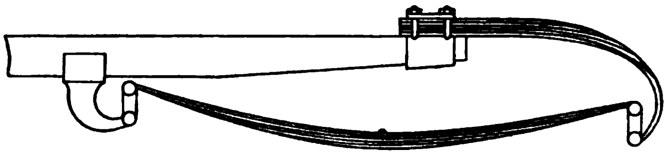


Fig. 275.

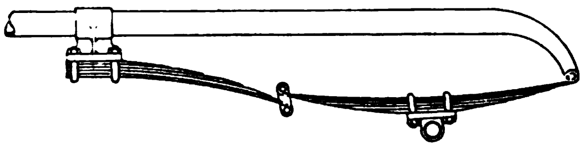


Fig. 276.

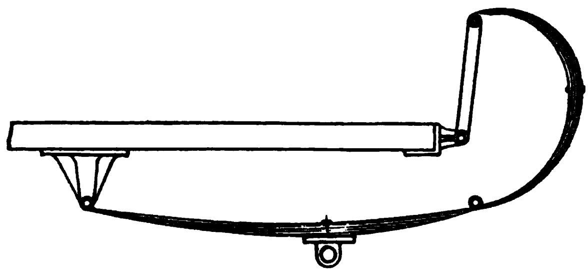


Fig. 277.

tage of taking all the weight off the frame at only two points instead of the usual four. This is a very small matter, and it has the advantage that one complete spring attachment is saved, which saves both weight and cost.

Another good plan is to have a cross back spring taking the ends of the two side springs. This is shown in fig. 216. In this case the frame is carried on three points well distributed. In arranging a car with this form of back spring there is a limit to the length of side spring that can be conveniently got in. The cross back spring cannot be carried much outside the back of the frame, and so the length of spring behind the axle will be determined by this. It is not good to have the car overhanging the back axle much, and this limits the distance for the length of the spring behind

the back axle. The part in front of the axle may be longer than the part behind, but should not be very much so, and, therefore, this limits the length of the side springs. Thus, if the frame overhangs the back axle less than 18 inches, the side springs cannot be much over 3 feet 6 inches long. On the other hand, the back spring will add to their effective length, so that, if the car has a standard gauge of about 4 feet 6 inches to 4 feet 9 inches and the side springs are well apart, the springing should be easy enough.

In some cases a single elliptic spring is used, but, to avoid an extension of the frame, the spring is extended in the way shown in fig. 275. This makes a good arrangement, but may put local stresses on the frame, which should be provided for. A variation of this, as used with front springs, is shown in fig. 276.

The old **C** spring has also been used and should make a very easy springing indeed, either with or without a cross back spring (fig. 277).

In front springs the single elliptic is very nearly universal, and seems to answer all requirements. It is common to put the front axle a little in front of the centre of the spring, as this shortens the frame and does not materially affect the comfort. Sometimes the front axle has been carried on a cross front spring, but this necessitates radius rods to it, and it has not come into general use.

In order to make a car ride comfortably, it is necessary that the axles should not only be able to move up and down, but also to tilt a little under the car. That is to say, if one wheel only goes over a stone or a bump, the car should not be tilted over to the same extent. In the case of the front axle this is fairly well provided for by the closeness of the springs. At the back, on the other hand, the springs are usually further apart in order to minimise the stress on the axle. In this case a back cross spring allows the axle to tilt easier. There must be some limit to the freedom to tilt, or else the car would have no stability. With a cross back spring and side springs the car is carried partly on the side attachments, and this gives it stability, and with the ordinary arrangement of front springs gives a very pleasant riding car, and one with ample stability. A cross front spring would give no lateral support to the car, and it might then be necessary to carry both ends of the back springs on the sides of the frame to get enough stability.

As the car will ride most easily if the axle is able to tilt underneath it, it might be an advantage to bring the back springs slightly nearer together when the cross back spring is not used.

Whatever be the form of the springs, they must be made of the right temper of steel in order that the car should ride easily. The great thing to be avoided is to have the springs too "springy." The elasticity should be such that they will take up most of the power put into them, so as to deaden it; if too elastic the car jumps with every bump in the road, so as to be most uncomfortable. This is most felt in cars with a short wheel base, as in them the occupant is pommelled by the back of the seat. It is possible to have comfortable short wheel base cars, as was proved in the old Locomobiles. These had a wheel base of only 4 feet 6 inches, yet were very easy riding, but they had very long easy springs. They were probably helped a great deal by the weight being almost all in the middle of the car, instead of being partly at one end and partly at the other, but the springs were the main point.

The same strength can be obtained from a spring in two ways; either by increasing the number of laminæ, or by hardening the steel. That is to say,

that for a given length and width of spring, the softer the steel the greater will be the number of laminæ (of a given thickness) required for a definite strength. The spring with the hardest steel and the least of it will be the most springy. Accordingly, the best carriage springs are made of very soft steel. Most of the French cars have evidently very soft springs, since, although very heavy in proportion to the load they carry, they have a lot of give in them. I believe that the best results are obtained by having the steel just as soft as it can be without continuously settling down. Carriage springs of the best makes are often so soft that they settle perceptibly at first. Springs made of a great many thin leaves deaden shocks better than those made of a few thick ones, owing to their greater friction. For this reason spiral springs are hardly suitable, as they have little internal friction.

The front springs are almost always put under the frame and in line with it. The front ends are fixed to the frame, and the back ones fixed to a shackle. In this case the shackle is often in tension, and carried on a bracket, much as in fig. 278. There seems no objection to having the shackle

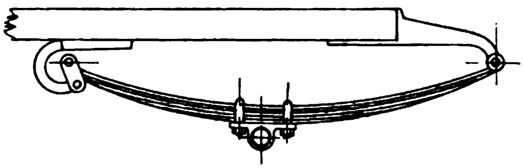


Fig. 278.

in compression, as in fig. 215, and this is cheaper and lighter, as the bracket is saved.

Springs vary so much in width, length, and number and thickness of laminæ that it is not possible to give a rule for their size; this can only be found by direct experiment. The hardness of the steel they are made of also has a very great influence. As an instance the following may be taken:—

A car weighing 18 cwts., and built for four people, had springs as follows:—Front springs, single elliptic, 3 feet 4 inches long, $1\frac{1}{2}$ inches wide, with four leaves $\frac{1}{4}$ inch thick. Back springs, single elliptic, with a cross back spring, side springs 3 feet long, $1\frac{1}{2}$ inches wide, and five leaves $\frac{1}{4}$ inch thick. Cross back spring same dimensions as side springs. The springs were tempered fairly hard, and gave pretty easy running, but were too jumpy. They were softened till quite soft, and three leaves were added to each of the back springs, and two leaves to each of the front ones. The result was that the springs were just about as strong as before—*i.e.*, gave just about the same amount under the same load, but the increase in comfort in the car was very great. It will be evident from the above that no formula can be given for the strength of springs, unless the exact nature of the steel they are to be made of is known. The above, however, proved very satisfactory.

Radius Rods.—The function of the radius rods is to keep the back axle in its right place, and to prevent it from turning. In chain-driven cars they take the strain of the pull of the chain, and in many cases keep the axle from twisting round when the brakes are put on. In live-axle cars they prevent the axle casing from twisting round under both the driving and braking strain.

Small cars have been made without radius rods at all, the springs

serving the purpose. In chain-driven cars there seems no reason why this should not be satisfactory if the front portion of the spring is fairly stiff, as there is no twisting moment on the axle due to the driving stress, this being direct from the chain. In this case the front end of the spring must have some means of altering its position fore and aft, generally by screw and nut, in order to adjust the chain. In live-axle cars the matter is not so simple, as there is a twisting moment from the driving stress. Further, it is not of very great consequence if the axle does twist round a bit in a chain-driven car, but with a live-axle car it throws the universal joints a good deal out of line. It can be shown by taking off the radius rod of an ordinary live-axle car that the amount the axle twists round is very considerable with any ordinary arrangement of flexible springs.

Possibly in small cars a satisfactory arrangement might be made by having the front portion of the spring pretty stiff, and having the back part very easy, say by a C-spring, as in fig. 277.

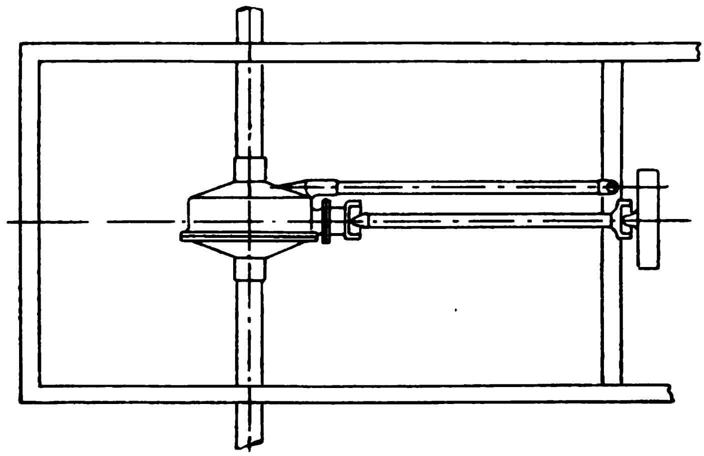


Fig. 279.

In a chain-driven car the usual arrangement of radius rod is that shown in fig. 216, and it has some arrangement of screw adjustment to take up the slack in the chain. Both ends of the springs are hung on swing links to accommodate this.

Practically all modern cars have some form of radius rod besides the springs.

In the case of the back axle, without any radius rods except the springs, the distance of the back axle from the gear box will slightly vary. This entails having a joint in the shaft to allow of a certain amount of end play. The amount is very small, but even in most arrangements of radius rods it is perceptible, and efforts should be made to render it as small as possible. If the radius rod was pivoted *exactly* on the centre of the forward universal joint there would be no end motion; but as this is generally impracticable, the endeavour should be to approach as near to it as possible.

The simplest form of radius rod is a strong tube brazed into the casing of the back axle and pivoted as near the centre of the forward universal joint as possible (fig. 279). The back springs in this case are hung on

shackles at both ends, and the radius rod is relied on to keep the axle square to the car, as well as to stop its turning round. Alternatively the radius rod may be used only to prevent the rotation of the casing. In this case it will be lighter and little, if any, more expensive if made of two tubes, as in fig. 280. The rod is here shown fixed to the casing by the same bolts that fasten the casing together. This looks like a make-shift, but it works satisfactorily. A radius rod so arranged is hardly strong enough laterally to keep the axle square to the frame, consequently if the springs are hung on shackles at both ends it may get slightly out of square. The amount it does so is so slight as not to be of great importance. On the other hand, if the springs are fixed in front they will keep the axle square, but there will be a little more end motion on the propeller shaft, as the centre of the front springs will not coincide with the centre of the front universal joint, and the front end of the radius rod must be carried on a swing link to allow for the end motion.

To avoid these difficulties radius rods are placed at the side to keep the axle square, and another in the middle to keep it from turning. This seems rather expensive and heavy, and it is simpler to arrange for the necessary end motion in the way described in the last-mentioned plan.

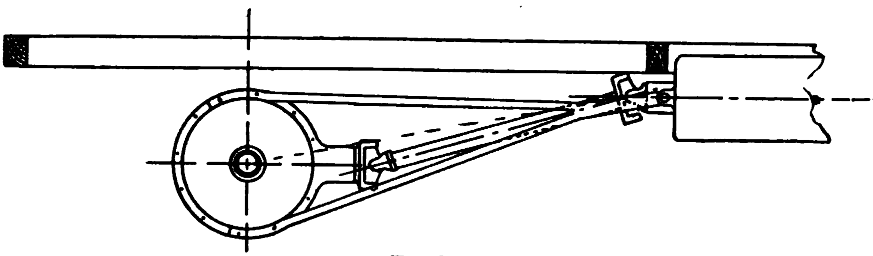


Fig. 280.

The spindle of the bevel gear is sometimes set parallel with the gear box as shown; but this is wrong, as it should be set as shown by the dotted line.

A plan which is coming largely into use is to have the propeller shaft run in a tube which also forms the radius rod. This makes a very neat arrangement, and does away with one joint. It also forms a very good support for the propeller shaft, and removes any possible objection to the pinion in the back axle being overhung. It requires some care in the erection of the car, as it is necessary that the bevel pinion shaft should lead *exactly* fair on to the gear-box shaft, but this can be easily arranged (fig. 215). In this case it is very common to allow the after bearing of the gear-box to take the stress on the radius rod. As the stress on the end of a long radius rod is very small, this plan works very well, but it is possible that it would be better to pivot the front end of the tube instead. This can be conveniently done if there is a cross piece behind the gear box as in fig. 215.

In this arrangement, as in the last, we may either have radius rods at the side, or trust to the springs to keep the axle square. Or we may pivot the radius rod in such a way as to be efficient by itself.

If a single tube is employed as a radius rod it should be of good substantial section, as the stress on it near the back axle is considerable. The stress is inversely as the distance from the centre of the back axle, and,

therefore, the radius rod should, theoretically, be tapered. The same effect is obtained by making the radius rod of two tubes, as in fig. 280, as these form a strut and tie.

In chain-driven cars the compression stress on the radius rod can easily be calculated, as it is the same as that on the chain (see Chapter xix.). The rod is a simple strut, the strength of which is calculable.

The twisting strain of the radius rods to the back axle can be readily calculated, as it is that which will make the wheels slip on the road. If a single tube is used, it is found that a fairly stout tube of large enough diameter to take the propeller shaft is strong enough. If the rod is of two tubes, as in fig. 280, the diameter is generally about $\frac{3}{4}$ inch and the thickness $\frac{1}{16}$ inch for a car of, say, 15 cwts., and proportionate dimensions for larger or smaller cars. These dimensions will vary somewhat with the length of the radius rods, as the longer a tube is the stronger must be its walls.

Radius rods must be so arranged that they will not prevent the axle tipping underneath the car. The axle can be, theoretically, restricted to an up-and-down motion parallel with the car, and the slack of the joints will often allow of enough tipping motion for practical purposes. Still on rough roads the amount that the axle will have to tip is very considerable, and should be arranged for, and if radius rods restrict this motion there will be some jolting when riding over rough ground.

In all cases where two universal joints are used, the axle should be so set that the line of the spindle of the bevel wheel points straight for the centre of the forward universal joint. Sometimes the spindle has been set horizontally, as in fig. 280, with the result that there is a great deal of unnecessary work on both the forward and after joints. The dotted line shows the amount of decrease in angle by setting the spindle in line.

The radius rods should also be so arranged that the spindle is in line under all circumstances. In some cases there have been drag links fitted to the top of the axle casing, in combination with radius rods at the side, the whole forming a sort of parallel motion. In this case, even if the spindle is set to point right in any one position, it will not always do so if the axle has any up-and-down motion on the springs.

Sometimes radius rods are fitted to the front axle in combination with a cross front spring. In this case they do not need any adjustment, and do not take any twisting strain. As they are generally in compression, they may conveniently be made tubular.

Brakes.—The brakes of a car must always be reliable, as upon them depends the safety of the occupants as well as of the public. They should be able to stop the car on any hill when ascending or when descending, however long the slope, and whether the car is going forwards or backwards; moreover, they should never become so hot as to lose their power. They should take hold easily and not grip too suddenly, or it will not be possible to use them on a greasy road without skidding.

The general arrangement of brakes is to have one worked by the foot on the countershaft and one from a hand lever on the back wheels. This plan does not seem capable of improvement. Some makers have made a point of having three brakes, but there does not seem the slightest advantage in this if the two brakes are reliable. One cannot possibly do more with any number of brakes than stop the wheels going round, and, as a matter of fact, the greatest possible braking effort is that which just does *not* stop the

wheels. A single brake in good order will prevent the wheels going round, so it is not clear what is the advantage of having three.

Each brake should be made capable of stopping the car on any hill if the wheels can grip the road, and should be so strongly constructed that there is no chance of its breaking down; and that it will run several thousand miles in a hilly country without adjustment. In fact, there is no more excuse for a brake failing than for a boiler bursting, for with sufficient careful attention to the construction there will never be a failure.

Different drivers use their brakes in different ways; the right way seems to be to use the foot brake in all ordinary cases for stopping the car, and the hand brake only for making it stand still when not occupied. The foot brake acts through the differential gear, and, therefore, when using it the strain is evenly distributed between the back wheels. This is necessary, as if one wheel has to do more than its share of retarding the car on a steep hill it is liable to slip a little, and then the tyre wears very quickly. Many side brakes have compensating arrangements to make the braking effect equal on the two wheels when the side brake is used, but they could only equalise it absolutely if the coefficient of friction was exactly the same on the two brake drums, which is not often the case. It is sometimes objected that using the foot brake "strains the parts" of the drive and differential. True, but we have got to the stage in car construction when it ought to be possible to make the differential and other gears, &c., to stand the strains that are put upon them. At all events, with gear-driven cars this ought to be absolutely so. With chains, as now made, there seems no reason why this should not also apply to them, but the effect of wear on chains is greater than on the gear.

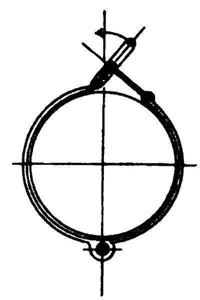


Fig. 281.

In any case, we have a brake on the wheels in case of any part giving way from any quite unforeseen cause.

All modern brakes are made to hold equally well in both directions. This is done by fixing the middle part of the brake band so that the lever tightens it up both ways. Fig. 281 shows a common arrangement. In this case there should be a fixed fulcrum for the lever, as shown, as this ensures that the brake shall be kept off the drum at both sides when it is off. The brake will hold quite well with no fixed fulcrum here; but when the car is running and the brake is not being used, there is nothing to keep it from touching the drum on one side, and this makes it rattle badly.

Many variations in the proportions of external brakes and their levers can be made, but there is not much difference in principle. Lately, it has become the fashion to a great extent to use expanding brakes, instead of external ones. There does not seem to be the slightest advantage in this in principle, as, for a given-sized drum, the friction will be much the same whether the inside or the outside piece moves. The great advantage in the expanding brake is that, as the drum is outside it, the dirt does not get in so easily as in the other form, and this reduces the wear. This is particularly the case with the back-wheel brakes, where there is liability for the mud off the wheel to get in. These can be arranged with the same system of levers as in the external pattern. A simpler and cheaper pattern, with very few parts to rattle, is shown in fig. 282. In this the ring is simply split, so as to

form the spring which releases the brake. The lever may be a stamping, in which the only machined part is the hole drilled in it.

In all the above figures the brake is shown as having the pull from the pedal or lever across the shaft. This will naturally be so in the back brakes, and in the countershaft brakes of chain-driven cars. In order to make it so in gear-driven cars a bell crank has to be used. A simpler arrangement is shown in fig. 283.

In the actual construction of brakes, it is very important that they should be metal to metal. Malleable cast-iron drums and cast-iron brake-blocks seem to be the best materials, but several other combinations work well. Gunmetal brake-blocks and malleable-iron drums work well. Nothing seems to work quite as well as cast iron on cast or malleable iron, which are the cheapest possible materials when renewals are required. The brake-drums may be plain, flanged, or V-grooved drums. The plain drum seems best, as the brakes are less liable to rattle from there being side play when the pins get a little worn. The V-groove increases the holding power considerably.

The most important and most frequent defect in brakes, and the cause of nearly all the difficulties with them, is that they are made too small.

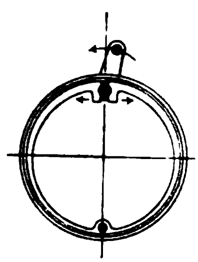


Fig. 282.

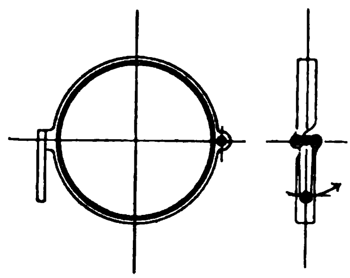


Fig. 283.

England is a somewhat hilly country, with many winding roads, and a considerable traffic; consequently the brakes are in constant requisition, and, therefore, subject to much more wear than in a flat country with straight roads and little traffic. Brakes should be usable for a reasonable time without adjustment or renewal; for if constant adjusting is needed, they are almost sure to fail at some critical moment, and lead to serious results. They should be usable on any hill without getting seriously hot. Small brakes will work, if kept cool by pouring water on them, or in some other way, but this is an unnecessary complication, as the brakes should be large enough to do their work without becoming unduly hot. Whether the brakes are expanding or external will have very little to do with either the wear or the heating. What is wanted is plenty of surface and good diameter.

There has been a very great increase in the size of brakes in most of the best makes of cars of recent years, but there is still a tendency to make them far too small in some cases. The best practice seems to be to allow 2 to 3 square inches of surface on the drum of the foot-brake for each hundredweight that the car weighs when loaded. The back-brake drums are each made about the same size as the foot-brake as a rule. In fact,

there are great advantages in making the side brakes and back brakes interchangeable.

In many cases brakes are made larger than is required by the above rule; if this can be done without increasing the weight too much it is certainly an advantage. It is quite easy to make small brakes hold perfectly well when thoroughly adjusted, and, in fact, quite small brakes can be made to stop the wheels, but they will want constant adjusting. A car with brakes that want adjusting every three days and renewing every three weeks in a hilly country, as some small brakes do, is not at all satisfactory, even though they may be quite powerful enough to stop the wheels when well adjusted for show purposes.

Wheels.—Wheels for cars are either of the wire bicycle type or the wooden artillery type. For bicycles and tricars the wire type is used almost exclusively, but for the larger types of cars the wooden one has very largely superseded it. There is no doubt that the wire wheel can be made to stand perfectly on fairly large cars, if made properly and heavy enough; but it is very doubtful whether it can be made any lighter for the same strength. There is no doubt that in theory it is so, and in some cases tests have been made showing that it is; but laboratory tests are of very little use in a matter like this. Wood will, in many cases, run safely with a much smaller factor of safety than metal, and this is probably true for wheels. Experience on the road over long distances in continuous work is the only reliable test. Even for racing cars the wood wheel is used by the large majority of makers of long distance high-speed cars. For instance, in the Grand Prix, 1906, which was 750 miles, and in which the winner averaged 61 miles an hour, the very large majority of cars had wooden wheels, including all those which finished at all.

Other reasons for using wood are that wooden wheels are considered to impart a neater appearance to the car, and are much easier to clean. The latter is a very important point.

As regards first cost there is little to choose between wooden and wire wheels for large cars, and although wire wheels may be less expensive for small cars, the saving is probably very small.

The construction of an ordinary wood wheel is shown in fig. 284. The hub is conveniently made a malleable casting. The washer is often so made, but it is better to make it a stamping, as it can then be lighter. It may also be stamped accurately enough not to require any other machining than boring the hole.

The spokes are usually of oak, but sometimes of hickory or acacia. This type of wheel has proved itself most surprisingly strong for its weight, and, compared with an ordinary carriage wheel with a wooden hub, often looks ridiculously light for the weight it carries. Still, car wheels are sometimes made a little *too* light, and accidents have happened owing to their collapse under stresses which they should have been strong enough to bear. The weakest part of many wheels seems to be the spokes a little way from the hub.

As the stress in the spokes increases from the rim to the hub, the spokes are tapered from the hub to the rim. This taper should obviously be gradual the whole way from the hub to the rim, as the stress increases gradually from the rim to the hub. In many cases wheels have been made in which the spoke is practically parallel from the rim to very near the hub, and then tapers out very rapidly to fill up the space in the hub. This is obviously

not correct, and such spokes often break just at the end of the tapered part if any special stress from side slip, &c., comes on them.

The spokes in car wheels vary from very slightly over an inch to $1\frac{3}{4}$

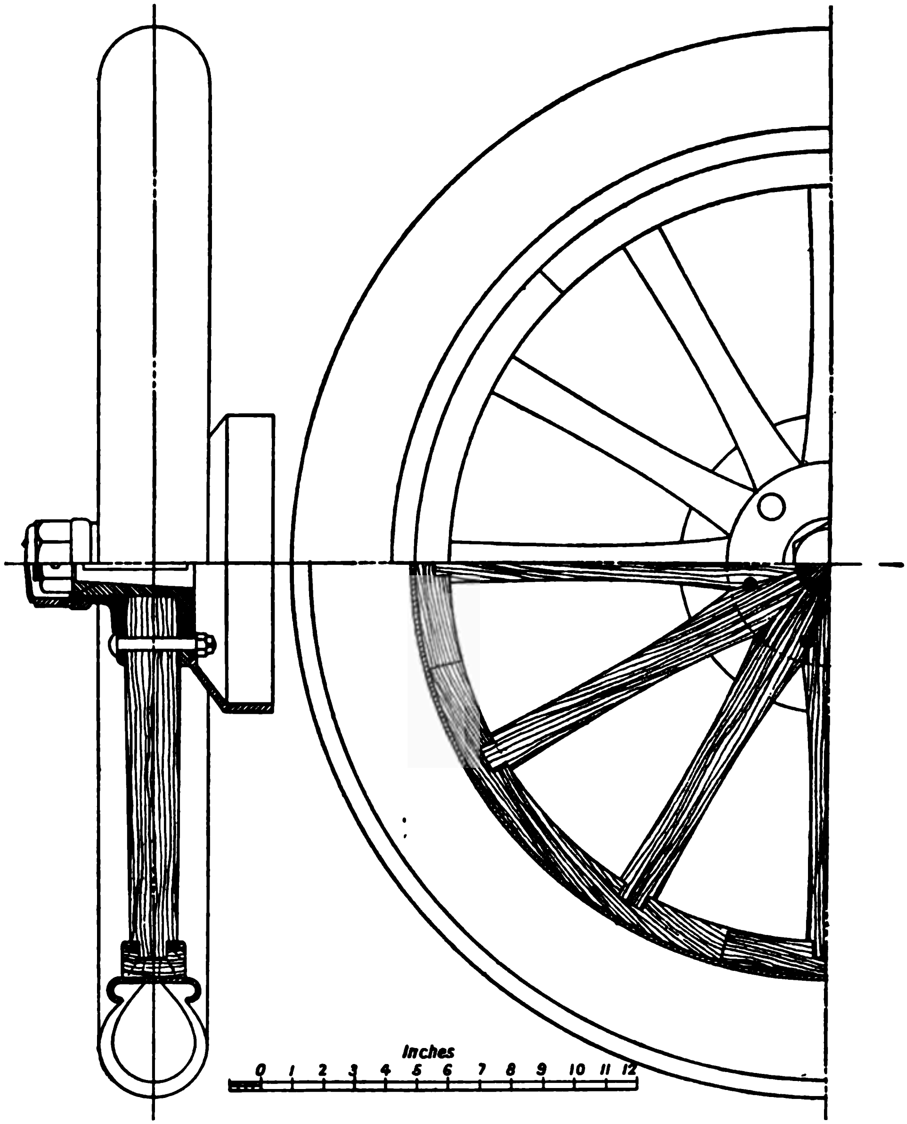


Fig. 284.

inches in thickness, and are generally from twelve to fourteen in number in each wheel. The smaller dimension is suitable for light two-seated, and the latter for large four-seated cars. The strength of the wheel will depend not

only on the size of the spokes, but also on the size of the flanges of the hub, and these might advantageously be made bigger than they sometimes are.

Where the car is chain-driven the chain wheel is often bolted on to the spokes. It then also serves as a brake drum. This does not seem very sound construction, as the bolt holes weaken the spokes, but it works very well if the spokes are big enough. If the axle is a live one, the brake drum is almost always bolted on to the hub, as in fig. 284. In this case the whole of the twisting strain goes through the hub and spokes. The chain wheel is often put on in the same way.

Which is the better it is difficult to tell. The latter is the neater, and probably the cheaper, as the same bolts that bolt the wheel together can hold on the brake drum. There seems no reason whatever why one should be the best for chain-driven cars and the other for live axles, as the maximum strains are due to the brake in both cases.

Wheels of lorries for heavy work are usually made of steel plate on the plan of traction-engine wheels. Sometimes they are cast in one piece of steel, which is cheap, but probably rather heavy. If the wheels do not have rubber tyres, steel wheels seem an advantage, as the wooden ones are liable to be shaken to pieces by vibration, but if rubber tyres are used, wood is generally preferred.

CHAPTER XV

FRAMES—STEERING GEAR.

Frames.—The various parts of the car must be carried on a frame. This is of varied design and construction. It should be so designed that it will conveniently carry all the various parts so that none of them have to be attached to the body, &c.

Taking the general design of the frame first there are two ways in which frames may be divided:—

1. Those which have an inside frame to carry the engine and gear box as against those which do not.

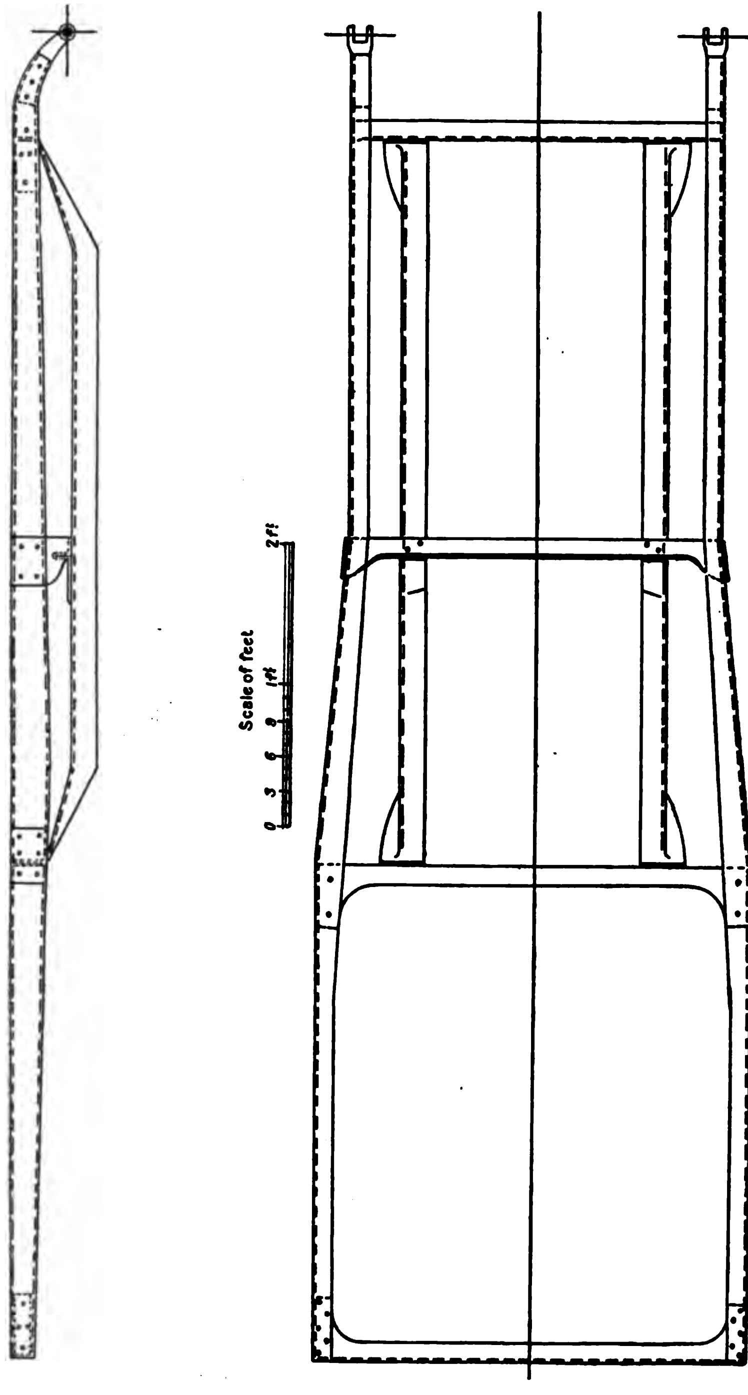
2. Those which have straight sides as against those which have the front part narrowed to allow the wheels to turn.

Fig. 285 shows a frame with an inside frame for carrying the engine, &c., and fig. 286, one without an inside frame. It will be seen that the number of joints in the former is nearly twice as great as in the latter, and that the cost of building the former will be nearly twice that of the latter. The weight of the frame will also be greater. Against this the inside frame makes a very convenient place whereon to erect the engine, &c.; while in the one without the inner frame the arms of the engine, &c., have to be extended to the outside frame. This is not a matter of great importance, and there is little doubt that the former is the more expensive of the two. As regards weight, the relative advantages are not quite so clear. The arms of the gear box and engine are necessarily a good deal longer if there is no inside frame, but there must also be cross pieces for carrying the inside frame, so the single frame arrangement will be the lighter of the two as the arms for the engine, &c., will be less heavy than the inside frame of the other arrangement.

The question of the narrowing of the frame in front is not so easy to decide. It has arisen from the lengthening of the wheel base and the necessity of making the front wheels turn through very large angles to allow of the car turning in a reasonable space. There is no doubt that a straight-sided frame will be stronger for its weight, as the bend in the frame is bound to weaken it. For cars of moderate wheel base there can be little doubt that the frame can quite well be made straight if the wheel gauge is of proper width, and still leave reasonable room for the body. This is probably also true for cars with fairly long wheel bases, as it is better to make the wheel gauge a little wider than to make the frame weak. The simplest way out of the difficulty would appear to be to make the frame straight, but of such a width that the front wheels can be put over as far as desired, and to let the body overhang the frame. It has often been assumed that it is impossible to make the body wider than the frame, but there seems no reason why it should not be so.

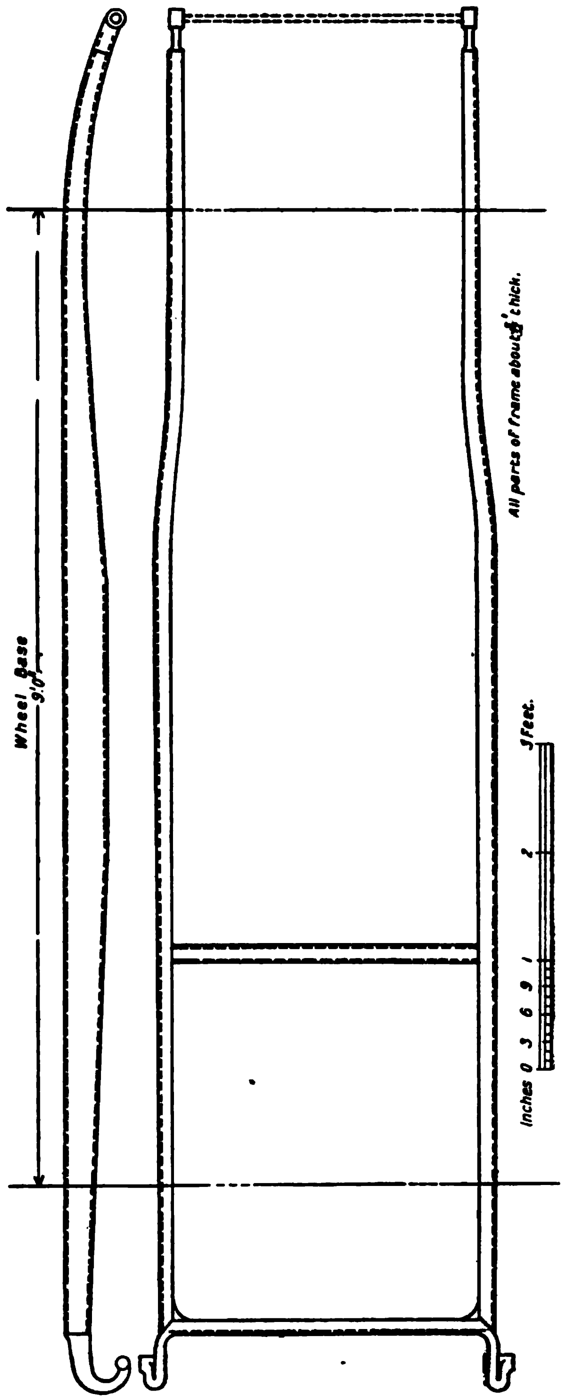
The straight-sided frame will undoubtedly be the lighter for its strength, and also, probably, cheaper.

The general arrangement in the chassis will be seen from figs. 215 and 216.



All parts of Frame about 5/32" thick

Fig. 285.



Wheel Base 9'0"
All parts of frame about thick.
Inches 0 3 6 9 1 2 3 Feet.

Fig. 286.

A practice is sometimes adopted of narrowing the frame in front very slightly, just enough, in fact, to be able to say it is not straight. This does not seem good, as it considerably weakens the frame without making the difference in widths enough to be of much use. The real object of narrowing the frame in front is to get it wider behind, in fact it should not be called narrowing in front so much as widening behind. It is obvious that it is not worth weakening the frame to widen the space for the body unless it is widened very perceptibly.

It will be seen that in Fig. 285 the frame is very gradually narrowed in front, the part between the wide and narrow parts being straight and with a cross piece at each end, while in fig. 286 the frame is narrowed suddenly. The former is undoubtedly the stronger, but it entails either narrowing the floor to the front seat or having it overhang the frame. The most ordinary construction is to narrow the frame suddenly, often much more so than in fig. 286, and if it is locally strengthened it is quite satisfactory.

Frames are occasionally raised behind, and are sometimes both raised behind and also narrowed so as to come inside the body here. The idea of this is to get the front seat and floor lower down. There does not seem much point in it, as, in the first place, it is not desirable in England to be too low, and, in the second, the height of most live-axle cars is limited by the floor of the back seat which must not touch the back-axle casing. It would be much simpler to raise the floor of the back seat without raising the frame, and every complication of the frame makes it heavier and more expensive.

The arrangement of the ends of the frame will depend on that of the springs. The front springs are almost always single elliptic, and the frame is carried out to take the front ends of them. Where single elliptic back springs are used, the back has to be carried out also. With a pressed steel frame the front springs are carried on the side members of the frame, which are carried out for the purpose, as shown in figs. 285 and 286. The back springs, on the other hand, are often carried on solid forgings rivetted to the side members. This hardly seems sound, as if it is right to make the side members with pressed steel of deep channel section, it would seem right to continue this section to the end of the back springs.

Where double elliptic springs are used such extensions are not necessary, and where a cross back spring is used it is generally possible to attach the cross spring to the back of the frame itself.

Materials.—The material most used for frames is "pressed" steel. That is, the parts of the frame are channel section, the depth of which varies according to the stress on that part, and the parts are pressed out of steel plate in dies. This is, theoretically, the best form of channel frame, for the reason that the depth can be graduated according to the stress. It also forms a very convenient construction, as the front and back can be carried out to take the springs without having special attachments fastened on.

The illustrations (figs. 285 and 286) give, approximately, the proportions of some of the lightest frames of this type in use, but most makers construct much heavier frames than this. The reason is not quite clear, as the lighter frames stand perfectly, if straight or nearly so. In fact, the majority of the pressed steel frames now in use seem to be heavier than the channel steel frames needed for carrying the same load.

The parts of the frame may be either rivetted together or welded. The latter, undoubtedly, makes the neatest job, but it is possible that it involves

a very much softer grade of steel, and that this is one of the reasons why some of the frames are so very heavy.

It may be mentioned that the difference in actual cost between pressing the frames in a die and hammering them up by hand is not very great, and one maker at least uses frames hammered up in preference to those pressed. The cost of making special dies for pressing them is very great, and will only pay for very large quantities, whereas the cost of making a template for hammering frames upon is relatively small.

The disadvantage of the pressed frame is that it may fail from local buckling, while the tensile or compressive strain per square inch is relatively low. In theory the deeper a girder is and the thinner its webs, the stronger it is for its weight, but there is a limit beyond which the webs get so thin that they fail owing to local cracking or buckling.

Besides this, it is a great defect of the pressed steel girder that there is no strengthening at the root of the flange. Fig. 287, which represents the flange of a rolled girder and a pressed steel frame, illustrates this. The resistance to buckling is much less in the pressed steel frame than in the rolled channel girder.

Fig. 287.

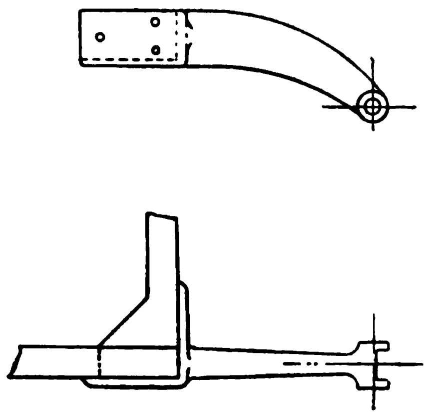


Fig. 288.

Another great disadvantage of the pressed steel is that it has a very low resistance to torsion. It is rigid vertically and fairly so horizontally, but not in torsion. This can, no doubt, be overcome by making it heavy enough, but then it is heavier than other forms.

A pressed steel frame of inverted U section, with the web at the top and both flanges downward, has been used. It is very neat, and is very convenient for fastening things to.

Channel steel is largely used for commercial work, and is far better in this latter respect. It might, perhaps, be more used in pleasure-car work with advantage, and has been so used with very good results by some firms. In fact, some of those who have used it have been quite satisfied that it could be made as light as the pressed steel, and have only given it up on account of fashion. One practical difficulty is to make a satisfactory arrangement for carrying out the side members to form the attachments for the front spring.

Wood has been largely used for frames, and is still used by some of the

best makers. It has considerable resistance to torsion, and is generally strengthened with a steel plate fastened to the longitudinal member to give it additional vertical strength.

The objections to the wood frame are:—

1. It is not at all easy to make the sides of the frame anything but straight.

2. The side pieces cannot be extended to carry the springs and make a neat job.

The wooden frame is usually constructed of ash in the form of a rectangle, from 2 to 3 inches square according to the size of the car, with a steel plate $\frac{1}{8}$ inch thick screwed against the side member to strengthen it. This plate is from 3 to 6 inches deep, and is generally flanged over at the edge to stiffen it. The joints are made with iron angle-pieces, and these are prolonged for carrying the springs, as shown in fig. 288.

Steel tube frames have been largely used for small cars, and seem worthy of more attention for large cars than they have had. They have excellent strength in every direction, including torsion, and, therefore, make a very

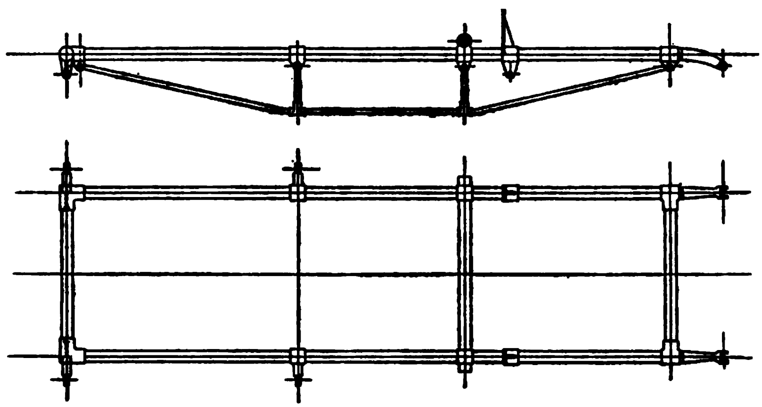


Fig. 289.

rigid frame. Probably they have suffered a good deal from having been used mainly by makers accustomed to bicycle work, who, finding that the ordinary sized tubes were too weak, tried to make them strong enough by making them thick, instead of making tubes of larger diameter. The strength of a tube for a given weight increases with its diameter up to a certain point, when it fails from local buckling. The proportion of diameter to thickness, which will give the best results for the same weight, has not been settled, but cycle makers have empirically adopted tubes having a thickness equal to about a thirtieth of the diameter.

Tubes are often used as the cross members of pressed steel frames to give them greater resistance to torsion, but it would seem that the proper thing is to have resistance to torsion in the longitudinal members.

Some of the pressed steel frames used in cars with very long wheel bases are now being trussed with a tie-rod underneath. It should be possible to make a frame lighter in proportion to its vertical strength if trussed than in any other way, but, in this case, the pressed steel section is not at all the best for the top member, as it becomes the compression member of a trussed

girder. In this case tubes would be, theoretically, the most perfect, and wood or channel steel better than the pressed.

Cars are now built, as a rule, with a long step running the greater part of the length of the car; in this case a very strong and light frame should be possible by making the step, and the brackets which carry it, the truss for a light channel or tube steel upper frame (fig. 289).

One point of some importance in the design of frames is the question of allowing for flexibility in them. A certain school of constructors prefer to have frames which are flexible, particularly in torsion, and to arrange that the machinery shall be so suspended on them as not to be disturbed by the movement. This seems to be really making the frame compensate for deficiencies of springing. There is no doubt that many early cars had such short springs that a flexible frame helped them slightly, but this seems to be an error. It would be better to have a rigid frame, and to let the springs be long enough to do their own work properly. If the frame has a material amount of give in it, there is always a great chance that the constant movement will ultimately develop weak places and cracks in it. Cars have rarely been in continuous use long enough for settling this point, but in the future there will be a demand for much more durable and reliable cars than those now made.

Steering Gears.—The Ackerman steering gear is used on all motor cars. The axle is fixed and the wheels are carried on pivoted arms at the ends; whereas in carriages and traction engines the axle is pivoted in the middle. The objection to the central pivot is that the strain on the steering gear is very great, if one steering wheel meets with more resistance than the other. For this reason traction engines have, for many years, all been fitted with absolutely irreversible worm and wheel steering.

In the Ackerman type the strain is much less, the distance from the steering pivot to the wheel track being much smaller. If the wheel touched the ground at the point where the line of the steering centre cuts it, there should, theoretically, be no strain on the steering gear from this cause. This can be arranged in three ways:—

1. The wheel may be sloped so that the bottom side of it touches the ground at this point.
2. The steering centre may be sloped outwards.
3. The steering centre may be put inside the hub.

All these have been tried but have not come into general use, although the wheels are often sloped inwards somewhat (see fig. 272). In practice there does not seem any advantage in this over vertical wheels.

The distance from the steering pivot to the wheel track can always be kept so short that there is no material strain from this cause, and, in any case, the steering gear has to be of reasonable strength in order to allow for side blows due to the irregularity of the road. The objection to putting the pivot inside the hub is that it makes the latter of excessive diameter, and even then the steering centres are very short.

It is quite possible that for heavy work the central pivot is the better type of the two, as the steering is easier when the vehicle is standing still. It is used in several of the most successful heavy lorries and tractors.

In order to allow of the wheels going over to different angles in going round a corner, the steering arms must be sloped outwards if they are in front of the axle, but inwards if they are behind it. It is necessary that they should go over to different angles, as the inside wheel has to describe a

smaller circle than the outside one. The method of setting out the steering is shown in fig. 290. It will be seen from the above that the exact angle is different for every different length of wheel base, and that in building cars from one set of patterns to different lengths of wheel base this will have to be considered.

If the arm P D connecting the pivot to the steering rod is set square to the car the steering wheel will have to be turned different amounts to the right and left for the same sized circle. This may be avoided by setting the arm as shown. If it can be arranged, there are considerable advantages in putting the rod connecting the arms behind the axle instead of in front of it. It is less liable to damage, and is out of the way of the starting handle. Also, it allows of the steering pivots being brought out nearer to the wheel track, as, if the arms are in front, they slope towards the wheel instead of away from it. The steering arms should not be very

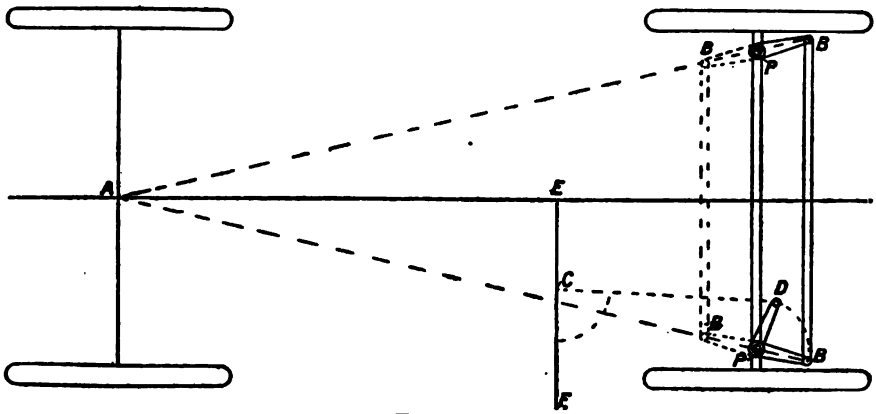


Fig. 290.

Full lines show steering arms and rods in front of axle.
Dotted lines show steering arms and rods behind axle.

A, Centre of back axle.
P, Steering pivots.
B, Ends of steering arms.
A, Line through P B produced should come to A.
C D, Rod connecting steering gear and arm in front axle.
Angle D P B should equal angle D C F.

short, as this makes the strain and consequent wear on the pins greater, while the effects of wear are of more importance in disturbing the steering. They should also clear the spokes well, or the slightest damage will make them cut the spokes badly. This fixes the place of the steering pivot if the arms are in front. The difficulty in getting the arms behind the axle is that, unless they are placed very low, the rod generally strikes against parts of the engine.

There are other arrangements which give more perfect correction for the angles of the wheels in going round a corner, but they necessitate more joints, are only superior when quite new and the joints are quite tight, and are both complicated and more costly.

If the arms are behind the axle the rod connecting them is normally in compression, but when in front in tension. This rod, in any case, is

generally made with double eyes at the end, either screwed or brazed into a tube or a rod. In some cases, two flat bars are used instead of a double eye, and then they simply want drilling to distance. This is probably the cheapest form of all. It is rather doubtful whether it is an advantage to have the length of this rod adjustable or not. It is certainly a convenience in setting the wheels, both when erecting and in case of damage; but, on the other hand, there is always the chance of its being adjusted wrongly by an unskilled repairer. For good wear all pins should be of ample size, and all joints be cased in leather and filled with grease.

In the older cars the steering was worked by a tiller pivoted at the front of the car. This is mainly restricted to very light American cars. It is made to lift up, so as to be out of the way when persons are getting into the car.

If tiller steering of any kind is used it is much safer to work it the reverse way to the usual former practice, which was to move in the opposite direction to that in which the car is to turn. The result of this is that, as the car goes to the right, centrifugal force throws the body to the left, so that the tendency is to "put the helm harder over." If the steering is reversed, as is done in one of the most successful tiller-steered cars, this tendency is avoided. Wheel steering is so universal that probably it is not worth while, commercially, building anything else.

This is now almost invariably of the "irreversible" pattern. That is to say, it is worked by either a screw and nut, or worm and wheel. It may be rather doubted whether this is really necessary for small cars, which do not go very fast. Many of the early ones were steered simply by having an arm at the bottom of the wheel spindle and using it as a tiller, which was quite satisfactory up to 30 miles an hour. This is necessarily cheaper than any "irreversible" form, and appears perfectly good if the steering centres are sloped rightly.

As a matter of fact, most pleasure cars do not have an irreversible steering. Traction engines and lorries have; that is to say, the pitch of the worm and wheel and their diameters are so arranged that no possible pressure applied to the wheel will make it turn the worm, as the teeth will break first. This makes a very slow steering, but cars require a quick one with quite different proportions, so that the wheel can turn the worm quite easily. This is evident, as the front wheels can be pushed over with one's hands, and the steering wheel turned quite easily. In fact, this is the usual way of putting the helm hard over when pushing a car about in a garage. Irreversible steering, therefore, now only means that the gear has so much friction in it that the front wheels cannot easily affect the steering wheel.

Although it is now usual to have a worm and wheel or screw and nut, a bevel pinion and quadrant or a rack and pinion have both been used with quite as satisfactory results, and are possibly cheaper to make. The difficulty is in casing them in, more especially the latter, but as there is not so much sliding friction the wear, with equal casing-in, should be less.

The general arrangement of worm and wheel is shown in fig. 291. In this the casing is generally split vertically, and the arm at the outside must be fastened on. The "wheel" is practically a sector, which is often bolted on as shown, and in some cases the arm it is bolted on to is also a loose piece keyed on the spindle. This means a great many joints, which add to the expense, and are liable to become loose. The advantage of having the sector loose is that it can be renewed when worn without renewing the rest of the parts, and also it may be made of gunmetal, if desired, so as to wear well

with the steel worm. It seems doubtful whether it would not be better to make some of these, if not all of them, in one piece, and this can be done by splitting the casing the other way, along the dotted line (A B). This will also probably make the casing cheaper to machine, as the facings can be turned at the same setting as the holes for the steering pillar are bored. If

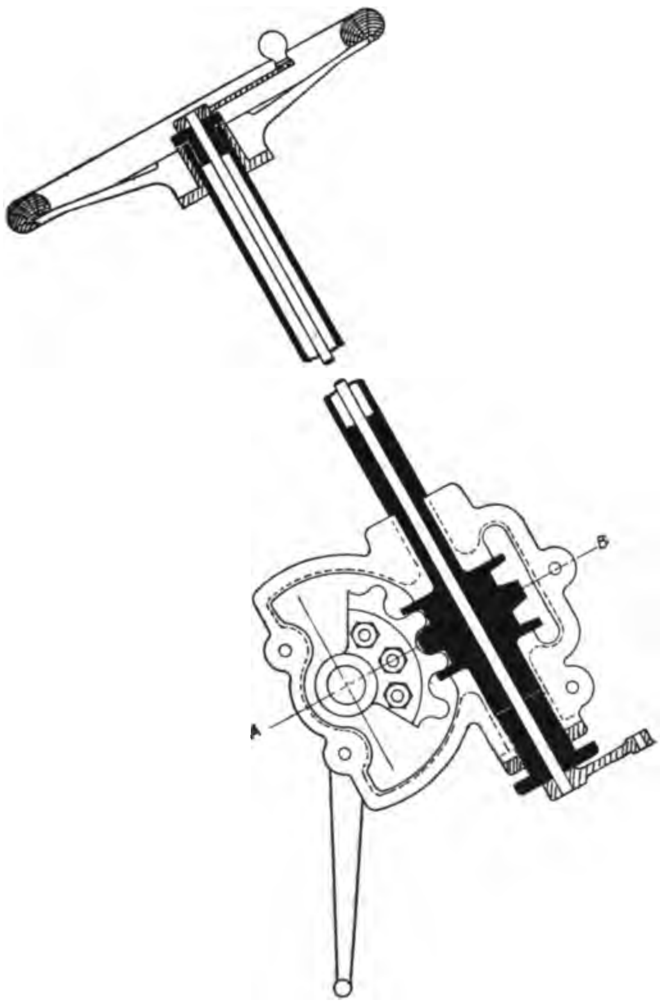


Fig. 291.

the sector and worm are hardened they wear so long that the extra cost of replacing the whole part would not be very great. On the other hand, if the sectors are made loose, they can be made as complete wheels and cut up afterwards, which will be cheaper than cutting each sector separately. The spindle and both arms might quite easily be made together with advantage, even if a loose sector was used.

In these steerings the worm has to be larger in proportion to that of the wheel (or sector) than is the case with those where a great reduction is made, as, otherwise, the thread is liable to acquire too great an angle. The proportions are generally about those shown; but it would be better to make the diameter of both worm and sector greater if possible, as this means less pressure on the teeth and less wear, and also the effect of wear on the steering is not so great. In moderate sized four-seated cars the worm is about 3 inches in diameter.

If the sector is a loose piece, it must be secured with bolts or rivets of good diameter, well locked; as small rivets are apt to work loose and shear.

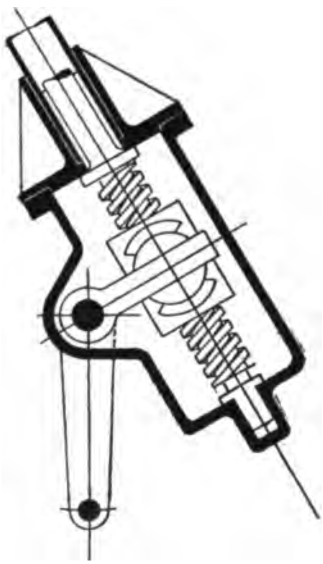


Fig. 292.

As shown, the worm and wheel pattern has a set screw and nut at the bottom of the casing to take up any wear. If this is provided, some really good means should be used to lock the nut, since if it works loose and screws itself up it locks the steering, and may cause a bad accident. It does not seem at all necessary to have such an adjustment.

Fig. 292 shows one arrangement of the screw and nut. This is capable of more variation of form than the worm and wheel, as the connection between the nut and lever may be made in several different ways. The variations do not, as a rule, much effect the cost or efficiency.

Comparing the nut and screw with the worm and wheel, the latter should be the cheaper, as it has the fewer parts, and it also has the fewer joints to wear and get slack. On the other hand, the screw has a much longer bearing on the nut than the worm has on the wheel. The worm and wheel is the most usual, and seems the best.

The ratios of the steering are convenient when it takes about half a turn of the steering wheel to put the steering hard over.

The arm on the steering gear is connected with that on the front axle by a tube with a ball and socket joint at each end, as in fig. 293. Occasionally springs are put behind the parts of the joint to give a spring drive to the steering, but are quite unnecessary; it seems better that the front wheels should follow the motion of the steering wheel without any springs to allow them to lag. The rod may be made with two joints at each end instead of the ball and socket; or, if the rod is long, probably one joint would do, but the ball and socket is not expensive, and is very satisfactory.

The wearing parts of the joint should be hardened and filled with grease.

In most cases the steering wheel carries some kind of a lever at the top to control the engine. Fig. 292 shows this, but one lever only is shown. This should be enough to control any modern engine, but there is no difficulty in fitting more by having them on tubes which fit one inside the other. The levers in this case do not go round with the steering wheel, and if it is desired to have a quadrant to hold them it must be carried on a tube fixed at the bottom end.

It is often preferred to make the control lever revolve with the steering wheel. In this case there must be some arrangement for giving an end motion to the rod through the steering column, and this must work a bell-crank lever through collars at the bottom end. The rod and column may be screwed into each other with a coarse-pitch screw, or the control lever may be held against a spiral sector on the top of the wheel by a spring. This plan has to be adopted if there are two control levers on sleeves inside each other, as otherwise moving the outside one would affect the inside one.

Sometimes the motion from the control lever to the engine is taken down the steering column by a flexible wire with a spring at the other end. This does not seem nearly as satisfactory an arrangement as having a solid rod and arm, as the wire is always more or less liable to break, and the spring is a complication.

The simplest and best plan is probably that shown.

The steering column itself is generally a tube, while the worm is made in the solid with the spindle that carries it in the case of the steering gear.

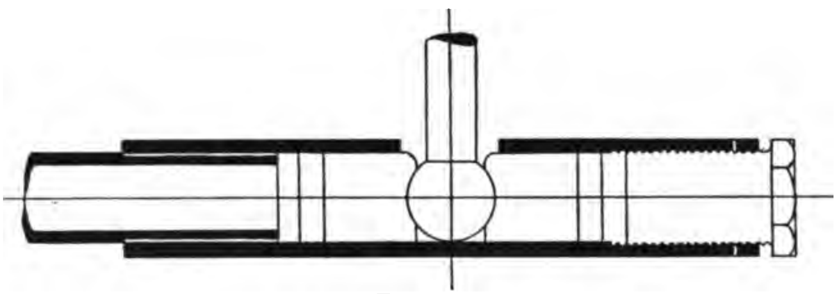


Fig. 293.

The column is often cased with polished brass, but it is doubtful if this is an improvement, as brass is apt to blacken clothes which touch it, however clean it is kept.

The wheel itself is generally a casting of aluminium, with four arms and a rim, and the rim is cased in wood; the diameter is about 15 inches. If this is used, the wood should be so arranged as *entirely* to case the rim, so that the hands cannot touch the latter. If the rim is of bent wood, and simply fixed to the spokes without the use of a metal rim, it is much warmer to the touch in winter.

The slope of the steering column is a matter largely of individual taste. The older cars had it nearly or quite upright, while the tendency now is for the column to be raked more and more, largely because the racing cars have it sloped in order to bring the steering further forward and nearer the front axle, for purely structural reasons. Cars have been built with the steering spindle quite horizontal, like a ship's wheel, but this is not found so convenient to steer with as one that is moderately sloped. An angle of about 30° with the vertical seems generally suitable. One point that should be noticed with steering spindles that are nearly horizontal is that in the case of any accident they may be very dangerous. If the steering spindle is fairly upright, in case of collision it bends over when the driver is thrown against it, and breaks his fall. If it is nearly horizontal, it will probably run through him and kill him on the spot.

CHAPTER XVI.

RADIATORS—DASHBOARDS AND MINOR ACCESSORIES—LUBRICATION.

Radiators.—The water from the cylinders is cooled by passing it through a "radiator." The object is to cool the water, and to minimise the weight of the cooling apparatus consistent with its being strong enough and not liable to leak.

The early radiators were pipes, some ¾ inch in diameter, with gills on them. This formed a fairly efficient cooling pipe, but occupied a great deal of room relatively to its cooling power. As engines became more powerful something more compact was needed, and other forms were produced. The simplest of these have tubes nearly as large as the older ones, but with an increased number of gills. This provides a much larger cooling surface for a given space, but air has to be forced through it. Another plan is to make the pipes themselves very small. This is an advantage, as less water can go along the middle of the pipe without coming in contact with the sides. Sometimes the pipes have been made of a flattened section. These have been made both with and without gills, and in each case cool well. Instead of gills, coils of fine wire have been soldered to them. This appears to combine the maximum of cooling power with the minimum of weight. A variation of this plan is to solder wire gauze round the pipes. One disadvantage of these is that the wires are liable to be knocked out of shape so as to be unsightly.

Another class of radiator altogether is that known as the "honeycomb." In this, instead of coils of pipe with gills, there are a series of plain pipes with no gills, but the water is outside the pipes, and the air goes through them. There are several ways of making these. In some the tubes are fixed in a regular tube plate. This is not often done. Usually the tubes are larger at the ends than in the middle, and they are just laid together and soldered. Sometimes the tubes are parallel, and have a ferrule slipped over the ends to make them bigger. Sometimes they are reduced in the middle. In any case, the radiator and tank are combined, so that there is no other water tank. In many of the radiators of this type the tubes are so close together that the water between them is a mere film. In others the tubes are so much reduced in the middle that there is a good thickness of water between the tubes.

In this type of radiator the number of tubes and the extent of surface in a given space will increase in proportion to the smallness of the tubes; at the same time there will be more joints and increased weight. To make this type efficient, it should be of large section and shallow. That is to say, there should be a lot of tubes, but they should not be very long. If the tubes are long the air in contact with them will be heated, and there will be a film of hot air in contact with the tube, and cold going through the middle of it. Also, it will take more power to make a good draught through the tubes.

In comparing the honeycomb type of radiator with the gilled tube, it would seem that the latter should be the lightest. This is because the gills have the air on both sides of them, whereas the tubes of the honeycomb have it on only one. On the other hand, the honeycomb divides the water into very thin films, so that it is bound to come into contact with the cooling surface. This can be done in the tubes by making them flat or of small diameter.

A great disadvantage in the honeycomb is the enormous number of soldered joints, often many thousands. These occasionally give trouble by leaking, and are then very difficult to make tight. The greatest advantage that they seem to have is that the radiator and tank are all in one. This makes a very convenient arrangement, as it makes the pipes very simple and also avoids having a tank anywhere else. The top of the radiator really forms the tank, and this should not be made too small. The smaller it is the lighter no doubt; but, on the other hand, it means that the slightest leak lets all the water out, and the result is the stoppage of the car.

Gilled pipe radiators are now frequently combined with the tank, so that this advantage of the honeycomb has disappeared.

If the gilled pipe radiator and tank are all combined, there are two ways in which this can be done. In the one, the pipes form a coil and the tank is round this. In the other, the pipes are all vertical and there are practically two tanks with the pipes connecting them. The latter has the advantage that it does not require so much power to force the water through the pipes, so that there is not so much need for a powerful pump. In fact, in many cases the pump is entirely dispensed with. It has more joints that may leak if badly made, but, of course, nothing like as many as the honeycomb.

Although the combined tank and radiator has come into very general use there are still adherents to the older system. The great advantage of the combination is that it makes the pipe arrangement much simpler and cheaper. There are fewer joints to leak or connections to go wrong. On the other hand, unless the tank is of fair capacity, anything going wrong with the water circulation is a much more serious thing. With a tank of ample size progress can still be made by filling up and running till the water is exhausted. With some of the combinations, however, there is such a small reserve space that this is almost impracticable.

In all the larger cars it is now usual to put some kind of a fan to suck the air through the radiator. Otherwise, the water is liable to become hot when the car is standing still or going slowly up hill. The most usual plan is to have a fan just behind the radiator, as in fig. 215. In some cases, however, the whole of the bonnet is made more or less air-tight, and a fan is placed in the flywheel. This seems to work very well in practice, but it entails a rather large flywheel, which, in consequence, has to run very close to the ground. This and the draught from the fans will tend to make a great deal of unnecessary dust. On the other hand, the cost of the separate fan and belt is saved.

In the matter of position, the radiator is nearly always placed vertically in front of the bonnet. If the tank is combined with it, this seems the most satisfactory position. If the tank is separate, it seems very doubtful whether it is not better to put it below the bonnet. In this case the engine is made far more accessible, as, when the bonnet is lifted up, all the parts can be reached. The position of the radiator seems to have been very much

affected by fashion, which has followed the arrangement of the latest racing car.

Sometimes the radiator has been placed at the side of the engine. This was done by Renault in his early cars. With a single-cylinder engine this was all right, as all the parts of the engine could be reached from the front, but with multicylindered engines it was given up. He now puts it *behind* the engine, and the air is sucked through it by a fan in the flywheel. This makes the engine very accessible, but the radiator adds to the length of the car and makes the bonnet look rather clumsy.

As mentioned, some makers dispense with a pump and trust to natural circulation. In very small cars this will probably be the best plan, as the cost of the pump and its connections will be saved. In larger cars it seems much more doubtful. If no pump is employed, a larger radiator will certainly be needed than if there is one. This will probably add more to the cost than is saved by doing without the pump. It will also add to the weight.

One point to be considered in radiators is that the types which require a very large frontage make the bonnet a great deal heavier. The honeycomb type will require a larger frontage than many of the gilled types. As mentioned, it is not much use making the honeycomb deeper, as there is nothing to break up the current of air. With the gilled-tube types, on the other hand, the air is well broken up, so that, in them, the radiator can be made deeper with advantage. The bonnet must be made the same size as the radiator, so that, if the latter can be made of smaller frontage and deeper, the bonnet can be made smaller and lighter. Radiators are generally dropped between the sides of the frame, as in fig. 215, in order to get more frontage, but even then they sometimes make very large bonnets.

Dashboards and Bonnets.—There is always some sort of a dashboard between the driver's footboard and the engine. This varies a good deal in height and is often kept a great deal too low. For driving about in England in all sorts of weathers, it is a very great advantage to have a dashboard high enough to be a shelter from the wind. Whether this looks nice or not is rather a matter of opinion, but it is certainly to be recommended. It is now very common to put a glass screen above the dashboard to keep the wind off.

A very good plan that has come into use to a considerable extent, is to recurve the top and sides of the dashboard. This forms a good shelter as well as a convenient place for small tools, &c.

Dashboards are generally made of wood, and, if they are to carry a number of articles, should not be made too light. Makers vary a great deal in this respect, as some of them practically put nothing on the dashboard, while others put a large number. Generally speaking, it seems that the less one has on it the better, but in small cars it is often the most convenient plan to carry both the petrol and oil there, as the tanks are not very large. Coils, if high-tension ignition is used, are also often put on the dashboard, as they are then easily accessible. From the dashboard forward there must be a bonnet to cover the engine. This must be easily detachable to facilitate access to the engine. One way of arranging this is to make the bonnet to lift off, but it is inconvenient to handle as a whole. Another plan is to fix it to a hinge at the back. This is very convenient, but entails a flat top to the bonnet in order to obtain a wide enough hinge, which is not always admired. In some cases the top hinges up and the sides hinge down.

Perhaps the best way is to have the bonnet hinged along the middle. In all cases it should be possible easily to take the bonnet right off if required.

Bonnets are generally made of aluminium or steel sheet, and should be kept as light as possible, as a good deal of weight goes in these small items. There will not be a great difference in the cost of the different types, though the type with a hinge on the dashboard would probably be slightly the cheapest, as it has only one hinge and the others have two or more.

It is very important that the bonnets should be held down in such a way that they cannot rattle, as this is most annoying. Spring catches of some kind are a very good plan. These are very quick to undo and hold the bonnet quite tight.

The section of the bonnet is usually determined by the area of the radiator. The tendency since the introduction of the honeycomb type has been to make the radiator of enormous area, and this necessitates a very large bonnet. This seems a great disadvantage, as it increases the weight of the bonnet a good deal without any advantage from it. It should be possible, with a gilled radiator, to make this of comparatively small section, but deeper longitudinally, and so make the bonnet smaller. This would lighten it materially, and also make a nicer looking car.

Petrol Tanks.—Petrol tanks are usually made of sheet brass or copper. They should be very carefully made so as not to leak. There should be absolutely no difficulty in this, but petrol tanks leak more often than is at all creditable to the industry. A good lap joint to the seams is necessary, and it makes it additionally secure to have this rivetted, as well as soldered. It is also necessary to have the petrol cocks and connections well put in, and these should have a nut on the inside of the tank and be soldered.

The great variation in tanks is in the position in which they are placed. In small cars, perhaps the best place is to put the tank on the dashboard. The amount of petrol that has to be carried is not very great, and a tank for this can quite well be put there. In this case, the tank should be kept high enough off the floor to allow the feet to be put underneath it. Otherwise, it will lengthen the car. Another place is to have the tank in the bonnet, or to make it a continuation of the bonnet. This is very convenient in most ways, but rather interferes with taking leads through the dashboard, if coils are placed on the dash. In both these arrangements the tank is conveniently placed and easy to fill.

Sometimes the tank is placed under the driver's seat in the body of the car, but here it lessens the room in the car to an inconvenient extent, and is very awkward to fill, while its low position hinders the petrol from running into the curburettor when the car is going up hill, unless the latter is put so low as to be very inaccessible.

A very common plan now is to place the petrol tank under the frame, and have this under pressure so as to force the petrol up into the carburettor. This has many advantages, as the tank is entirely out of the way and does not in the least interfere with any body that may be put on. Very commonly, however, the tank is placed very close to the ground, which is objectionable on account of the enormous amount of dust raised. If this position of tank is used, the tank should be put *in* the frame and not below it. One advantage of this plan is that the carburettor can then be placed as high up as desired, and there is no reason why it should not be placed level with the valves, as it used to be in some of the cars of years

ago. This very much shortens the inlet pipes and makes the carburettor easier to get at.

In this case the tank is often made round, which is, no doubt, the cheapest and lightest form, but if made rectangular it can be kept further off the ground, and there is no difficulty in making it strong enough without excessive weight. It is sometimes corrugated to make it strong and light.

The pressure on the tank is usually obtained by having a small pipe from the exhaust to the petrol tank with a non-return valve in it. This gives sufficient pressure for the purpose.

It is very convenient to have a gauge in the petrol tank to show how much petrol there is. A glass gauge can be used for this, but is not very suitable, the better plan being to use a float for indicating the level. Several such are on the market.

Steps and Mudguards.—The step to the front seat is often made a forging, with a flange whereby it can be fastened to the side of the frame. Steps seem often to be made very much heavier than necessary, and especially the flanges, which are often far thicker than the frame itself. The arm might often with advantage be made H-section and stamped.

In many cars, especially those with a side entrance, the step extends over the whole distance between the mudguards, as shown in fig. 324. It must then be carried on two or more brackets. It is generally made of wood, and covered with indiarubber matting. The tool box or the box for holding batteries and coils is often placed here.

Mudguards are either made of wood or of patent leather on a metal frame. The latter certainly look the neater when new, but the wood seems preferable on the whole. They should have a good width, and should come well over the wheels at each end. It is a mistake to have the mudguards so near the wheels that it is difficult to get the tyre off.

The back mudguard generally follows the wheel down to the step, which it joins either with or without a curve, according to how far the step extends backward. The front one, however, often goes far below the step, and almost to the ground. It is better to curve it in to the step, as this keeps it further off the ground and raises less dust, and requires one bracket less to carry it. In this case the guard should have a ledge along the inside to keep the mud from splashing the car body and bonnet. The brackets for carrying the front mudguards are generally attached to the frame, and those for the back ones to the body of the car. Care should be taken that these are as light as possible, consistent with being sufficiently strong.

Sprags.—Sprags are antiquated and useless for cars with good brakes, although they are still retained on many. When brakes could only hold cars from running forward, sprags served to stop its running backward when ascending a hill. There is no need for them for cars which have brakes that will hold equally well both ways.

The sprag has also been found useful in cars in which the foot brake and clutch are connected, as it is not very easy to start on a steep hill from the brakes. If the clutch pedal is lifted up gently the brake comes off before the clutch goes in, and the car immediately begins to run back. If, however, the brake is not connected to the clutch, there is no difficulty if the car has a proper amount of tractive force on its low speed, as it is quite easy to take the brake off as the clutch begins to grip. For these reasons a sprag on a car with proper brakes is a useless weight and complication, for, if the brakes

are too small, it would be better to put on brakes of a proper size than to carry about a sprag which will weigh more than the proper brakes.

The older form of sprag was simply a bar with a pointed end, attached to the frame; it ran along the ground at such an angle that it gripped when the car began to run backwards. It is a very unsatisfactory contrivance, as in the first place it has considerable weight, and in the second it does not always grip. Again, if the car ran back at all hard, it sometimes went over the sprag, which then prevented a forward movement until the sprag had been taken off and put on the right way again.

For these reasons a ratchet and pawl on the differential shaft is often fitted. This is better, as it is pretty certain to act, but if the pawl drops into place after the car has attained any material speed, the stress on the gear may be sufficient to break it.

The pawl is sometimes fitted with an automatic arrangement to pull it out of gear when the reverse is put into gear.

Should the Brake and Clutch be Connected?—The older cars almost invariably had the brake and clutch pedals connected, and also had the side brake connected to the clutch. In fact, a great many have this arrangement now. It is not quite clear what the advantage of this is, but, presumably, it was intended that it should ensure that the driving power ceased when the brake was applied. If the idea was that this made the car safer, as the brake would not be able to stop the car unless the driving of the car ceased, it seems with proper brakes to be quite unnecessary. Possibly with the early cars there was something in it, as they were fitted with very inferior brakes indeed. The brake drums were far too small and the surface also too small. Even now there are many cars which are not at all as they should be in this respect. With brakes of proper size, however, there is nothing in this point, as they will easily stop both the car and engine. Further, it is a perfectly natural action to put down the brake and clutch pedals at the same time without their being connected.

There are, on the other hand, many advantages in having the pedals not connected. A small one is that the engine can be used as a brake in case of partial failure of the latter. This is to say, the motion of the car can be made to run the engine while the full power of the brakes is applied. This may be in some cases an undoubted advantage, though brakes ought not to fail. In driving it is, however, a very distinct advantage to be able to put the brake on slightly, without disconnecting the engine, when there is much traffic. For instance, the car will make less noise when the clutch is left in and the slowing is effected with the throttle and brake than when the clutch is taken out. The greatest advantage, however, is in starting on a steep hill. If the brake and clutch are connected starting is sometimes difficult without a sprag, which may not hold properly, and is always an additional complication and expense. The difficulty is that if the pedals are connected and a pedal is allowed to make the slightest rise, the car immediately starts to run back, and by the time the clutch has taken hold the speed is too fast for the engine to pick the car up again and start it up the hill. Of course the foot may be lifted quickly and then the clutch will grip sooner, but this is not easy to do nicely. With the pedals disconnected there is no difficulty at all, as the clutch pedal is allowed to rise until it is felt that the clutch has begun to grip and then the brake is taken off.

If it was really desirable to only press down one pedal when driving, there would be no reason for having more than one pedal. Some cars have

been built with one pedal, which first takes out the clutch when it is pressed down, and then puts on the brake, but this plan has been generally discontinued owing to two pedals being more convenient.

The fact is that most people, in driving, even when the pedals are connected, put a foot on each when stopping, take out the clutch with the left foot and put on the brake with the right, even though pressing down the right only would take out the clutch. It is, therefore, evident that much more can be done with disconnected than with connected pedals. In addition, the connection is always more or less of a complication and expense, and in some cases may interfere with the action of the brakes.

The question of having the hand brake and clutch connected has other aspects. The hand brake is generally used for making the car stand still when unoccupied, or the driver is not in it, and the foot brake for all ordinary work. The former is, of course, also a reserve, and should be amply powerful enough to stop the car on any hill, should the foot brake fail. Under these circumstances there is no conceivable advantage in connecting it to the clutch. On the other hand, besides the disadvantages as regards the connection of the pedals, there is another, which is that a careless driver sometimes stops his car by the brake and not by disengaging the gear, consequently the engine continues running and the gear remains in gear, the clutch being held out by the brake lever. If any one then meddles with the brake lever, the clutch will go into gear and the car will start. Accidents of this kind occasionally happen, and in one instance the car went through a shop widow. If the brakes are not connected to the clutch this cannot happen, as the driver cannot leave the car till he has put the gear out of gear.

Pedals.—The foot brake and clutch are worked by pedals in two different ways—by pressing down and by pushing away. The latter has predominated, and may be accepted as the most convenient.

It may, however, be remarked that it is much more important to have the pedals the right distance from the driver with the push-away than with the press-down plan. Hence, it would be a great advantage if pedals were made easily adjustable to the most convenient position for any driver, whatever may be the length of his legs. The average length of the leg varies with different nationalities; for example, the average length of leg is much longer in Englishmen than in Frenchmen.

The pedals themselves can most conveniently be made stampings of H-section, and the general arrangement and form can be seen from the general arrangement shown in figs. 5, 6, 215, and 216.

The right-hand pedal should work the brake, and the left the clutch, this being by far the most usual arrangement.

Much care is required in the arrangement of the pedals on the car, in order to avoid unnecessary expense in erection. The best way, when practicable, is to carry them on brackets cast on the gear box, so that the whole can be erected ready to put on the frame at once.

Lubrication.—Good lubrication is essential for the successful working of all kinds of machinery, and generally every bearing has its separate lubricator, but this plan is not suitable for cars.

We may divide the question of lubrication into two parts—(1) that for the engine, and (2) that for the running parts of the car.

There are three main systems of lubrication in use for engines. The first is that common on most kinds of engines; in it there is a lubricator to each bearing, which has enough oil supplied to it to keep it cool. The second is

that in which the oil is simply put into the crank chamber and thrown over all the working parts by the cranks, &c., splashing in it; this is known as "splash lubrication." The third is that in which oil is forced under pressure into the bearings by a pump which draws the oil from the bottom of the crank-case, after it has flowed out of the bearings, and forces it through again. This is known as "forced lubrication."

The first plan is little used on motor cars, and is hardly suitable to them, as the numerous oil feeds require more attention than the driver of the car can give them. It is, in fact, a system more suitable to open than to enclosed engines, and may be used with these in marine work.

The splash system is very simple, and, in small engines, very satisfactory. There is no need of any complication of pipes or oil pumps, &c., all that is necessary being one pipe into the crank case with some means of feeding the oil. The various parts of the engine should be provided with good oil holes. In some cases the main bearings have had cups above them to collect any oil which might be thrown against the sides of the crank chamber, and to compel it to run into the bearing. The crank pin can be made with the bottom brass narrower than the top, so that the oil gets thrown against it and carried round into the top of the bearing, and the latter may also have an oil hole. The top end should have a good oil hole and oilway, and in this case seems to get an ample supply.

The difficulty in splash lubrication is to provide enough oil for the various parts, without having so much in the crank chamber that it works up into the cylinder and produces smoke. This can be avoided by carefully adjusting the quantity put into the crank chamber to the actual needs of the engine. In this case little or no smoke will be produced with engines of moderate power. This matter is of very great importance, as the making of smoke is a public nuisance.

The oil may be introduced into the crank chamber either by a hand pump, by which a charge of oil is pumped in periodically at such intervals as experience suggests, or by some automatic continuous means. The former is very simple and cheap, and it is practically impossible for it to go wrong. It may, of course, be forgotten, and in this case the engine will run dry, but this is a matter entirely in the hands of the driver. On the other hand, with all automatic feeds there is the possibility of their not working properly, and, as they are supposed to be automatic, they are trusted to, and this may lead to bad results.

Automatic feeds are of two kinds. In one the oil drips through a needle valve, which can be regulated to give the amount required; in the other it is fed by mechanical means. In the former case the oil may be fed by gravity, but is more often fed by the pressure of the exhaust. In this case the feed is automatic, as the pressure is caused by the exhaust, and when the engine stops the pressure soon ceases, and can be arranged to do so as quickly as desired. The pressure feed is probably much to be preferred, as the pressure is greater than in the gravity feed, and the drip is, therefore, more certain. Another great advantage is that the oil reservoir need not be above the drip, but can be placed where most convenient.

The great objection to the drip system is that the drip is apt to be irregular. The hole has to be so small, to keep the amount of oil down to that required, that the least variation in the thickness of the oil makes a great difference in the amount which flows through. This has been the drawback to the use of this system both for motor cars and other machinery.

The system of having the oil fed mechanically seems much better, and should be much more certain. In this plan there are two modes of application—viz., by forcing the oil in with a pump, or dropping it into a pipe with a bucket on a wheel. The former is the most certain, but possibly the most expensive.

One difficulty in splash lubrication is that, if the engine is inclined, the oil runs to one end of the crank chamber, and therefore that end of the engine gets an excess of oil, while the other does not get enough. If the engine is always going to work on a slope, as it may do if not set horizontally in the car, this can be remedied by having dippers on the connecting-rods of different lengths, so that they all touch the oil at the same level. In a car, however, the level of the engine varies; the great difficulty is that the oil runs to the after end of the crank chamber when the car is ascending a long hill. In the smaller engines, even in those with four cylinders, there is no serious trouble from this, as the crank chamber is so short that enough oil will splash from the after end to prevent any bearing becoming hot, especially if the bottom half of the crank chamber has webs cast across so as to keep some oil under each crank. It is also advisable to feed the oil in from the front end of the chamber, so that the highest point will get it first. If the supply is continuous, the oil cannot reach the after end until it has been first splashed over the parts of the other cylinders, and if it is by hand pump, a charge put in now and then up a long hill will insure all being right. There is no need to trouble about the supply downhill, as the engine is running quite easy, and, in fact, on a hill long enough for there to be any trouble, it is often stopped altogether.

The third plan of forced lubrication is coming largely into use for electric light and other enclosed engines. There is no doubt that this gives the most perfect lubrication which can be devised, as the bearing is always flooded with oil to such an extent that the surfaces do not touch. The objections to it in automobile work are:—

1. It is an additional complication and expense.
2. The oil has to be fed through several small pipes; should any one of these be stopped up, the bearing receives no oil and will get hot.
3. If too much oil is forced through the bearings, it may pass into the cylinders, and cause the engine to emit much smoke.

These objections apply to small engines more than to large ones, as the latter have larger pipes, pump, &c., and more room for them, while difficulty with splash lubrication is greatest in the larger engines.

Forced lubrication is usually effected by pumping the oil from the bottom of the crank chamber into the main bearings. A separate pump is not used for each bearing, but one pump delivers into a pipe with branches to the various bearings. The pump may be of any type, but a positive pump, such as is often employed for the water circulation, works very well. The oil issuing from the ends of the main bearings is caught in a ring and led through a hole in the crank web to the crank pin, and from there up a hollow in the connecting-rod to the gudgeon pin.

It will be seen that, while this provides a very complete system of lubrication, it requires to be very well installed to make it satisfactory. Should one of the pipes crack, with vibration for instance, the oil supply would cease, and the driver may not discover this until the engine has become hot and parts have seized. There should also be some efficient filter between the pump and the crank chamber to prevent dirt getting in and stopping

up the pipes. As there is only one pump, if the lead to one bearing was stopped up, the oil would simply flow into the others, and it would run dry.

As at present carried out by many makers, there is no doubt that the forced lubrication gives a very excessive amount of smoke, and wastes a proportionate quantity of oil, as compared with the older systems. If it is to be satisfactory, it must be so arranged that, while it gives a satisfactory supply of oil to the bearings, it does not force enough through them to work up into the cylinder, and for this reason it is not possible to run with anything like the pressure of oil usual in steam engines; in fact, there should hardly be any real pressure in the supply. In some cases, no doubt, an excessive oil supply is used because the bearings are so much too small for their work that, without it, they would be in need of constant adjustment; but the proper remedy for this is better designed bearings. If forced lubrication is used, the crank chamber must be provided with a well, below the level of the cranks, capable of holding enough oil for any reasonable run, and there must be means for filling this up to the proper level when required.

The lubrication of the various bearings in the car depends largely on whether they are ball bearings or not. In the former case no lubrication of the bearings is necessary, but it is necessary to lubricate the gear wheels in some way. If the bearings are so arranged that the oil will not work out of the gear cases through them, the best lubricant for the gears is probably very thick oil. In many cars with ball bearings, however, if this is used it works out through the bearings and makes everything filthy, and sometimes by oiling the brake drums it prevents the brake holding. The escape of oil from bearings may be prevented by using leather washers in the way described in Chapter xvii.

If oil cannot be used satisfactorily in gear boxes, grease must be used. This works very well though it probably does not reduce the friction quite so much as the oil would, as it does not flood the gear to the same extent.

In the absence of ball bearings, parts may be lubricated either with oil or grease. The most perfect lubrication is with oil continuously supplied, but is not applicable to all the running parts of the car. It is conceivable that forced lubrication could be applied to a live back axle, or to a change-speed gear box, in the same way as it to the engine, but the plan has not been tried, and would entail considerable complication.

On the other hand, it is not convenient to apply drip feed or any other system of continuous supply from one source to all the various parts, as the oil would have to be conveyed through many small pipes liable to be broken by the vibration of the car. This applies more especially to the bearings of the axles.

The only remaining alternatives are splash lubrication and the use of grease.

In the case of the live back axle and the gear box, the same plan may be used as for the engine—that is, to turn a certain amount of oil loose in the casing and trust to its being thrown on to the bearings in sufficient quantity to keep them oiled; but this is not satisfactory for the change-speed gear, as the oil in the gear box is liable to be charged with fragments of steel rubbed off the teeth in changing gear, and these, should they become embedded in the bearings, make them cut badly; a few steel chips sometimes suffice to cause them to be quite worn out in a few hours. If white metal is used instead of bronze for the bearings, the risk of damage is far less.

There are also difficulties in the lubrication of the live back axle. If oil is simply put into the gear casing it will generally lubricate the bearings within the casing all right, but the bearings at the outer end of the axle either receive so much that they cover the tyres or brake drums with oil, or so little that they run partially dry.

In the case of the gear box, the best remedy seems to be to lubricate the bearings with grease instead of oil, which is easily managed by having one large screw-down grease pot on the dashboard and pipes going from it to the gear-box bearings. It is not necessary to have separate grease pots to each bearing, if the pipes are of good size. This makes a very satisfactory plan, the grease entirely preventing the oil in the gear box from working out at the ends, and so favours cleanliness.

The same plan might be used for the live-axle bearings, but as the latter has motion with regard to the car on the springs, the pipes would be very liable to break. It is more usual to fit each of the end bearings of the axle with a grease pot which can be screwed down occasionally. This will keep the oil in the casing and allow of the inner bearings being lubricated with oil.

It is one of the properties of grease as a lubricant that it does not require such frequent renewing as oil fed in the usual way. This is a great advantage, as grease pots will seldom need to be screwed down during the course of an ordinary run, and an occasional screw down while the car is standing will generally be sufficient.

The front hubs sometimes have an oil well, but are almost always run with grease. The usual way of doing this is to have a loose cap at the end of the hub, as in fig. 267. This is occasionally taken off and filled with grease, which is forced through the bearing when the cap is screwed on again. The bearing will then run some hundreds of miles without attention.

It seems rather a defective arrangement to have to lubricate such a small machine as a car with two different lubricants, but in practice the amount of grease required is so small that there is no difficulty in carrying enough in the lubricators for a fairly long tour, and no trouble is experienced in this respect. The fact that they will run without lubrication is, however, a distinct advantage of the ball bearings, and is no doubt the main reason why they have come into favour for many parts.

No serious attempt has ever been made, as far as I am aware, to lubricate the motor car engine with grease. This has been done with other classes of engines with good results, and, if practicable, might be both convenient and cleanly, while smoke would be avoided.

Lubricators should be of ample size, so as not to require refilling at short intervals; be so placed that they can be filled without spilling the oil about; and the filling holes of oil reservoirs should not be inconveniently small.

CHAPTER XVII.

BALL BEARINGS.

Ball Bearings.—The question of fitting ball or plain bearings to motor cars has been much discussed, and very exaggerated statements have been put forward on both sides. The matter appears to be one which can only be settled by trial and error, like most things, but there are certain matters which can be discussed with advantage. It is, perhaps, rather unfortunate for the ball bearing that its advocates have pressed its advantages with little regard to facts, and have, in many cases, based their argument solely on its success in bicycle construction. They have repeatedly stated that the reason why it has come into use on bicycles, and not in general engineering, is that in bicycles the rider feels the difference in friction between the ball and plain bearing, whereas engineers generally are perfectly ignorant on the subject of the friction of their machines and indifferent to its amount. As a matter of fact, there is probably no machine about which there is less accurate knowledge than the bicycle, and most engineers have far more accurate records of the relative economy of different methods of construction than bicycle makers have.

There are two quite different considerations with regard to bearings. First, the amount of friction absorbed by them, and, second, the reliability combined with durability.

Friction.—With regard to the first, it would seem that the question of relative friction of a ball or plain bearing depends *entirely* on how well the plain bearing can be lubricated. The ball bearing is more or less independent of lubrication, as the balls take the place of the lubricant. In a perfectly lubricated plain bearing of suitable size for the work it has to do, the surfaces of the bearing never touch, as the oil keeps them apart. If, however, the lubrication is not good the surfaces touch and then the friction increases. The effect of this is very well seen if the load on any bearing is increased until it will not carry any more. Up to a certain load the friction does not increase materially, but there comes a point when the load squeezes the oil from between the surfaces, and then the friction increases very suddenly and the bearing usually seizes. The pressure which a particular bearing will carry varies greatly with the material it is made of and the nature and means of supplying the lubricant. The lubricant which will give the least friction with a light load and high speed is not that which will carry the greatest load without seizing at low speeds, the former generally requiring a comparatively thin oil which flows freely, and the latter a thick oil with a great resistance to being squeezed out of the bearing. Moreover, the thicker the oil the less of it is wanted, as it stays in the bearing longer without running out at the ends. When the bearing runs in an oil bath the friction is very small indeed, so small that all the bearings of a car would absorb but a small part of the power spent in driving it, especially at high speeds and up hill.

From careful tests it has been found that, when run in an oil bath, the friction of a bearing is about $\frac{1}{1000}$ of the load, but this is not a con-

dition which can be often fulfilled. There is no reason, however, why it should not be in the case of the engine bearings, as all are closed in and the oil may be circulated through the bearings by a pump if desired. In this case, with bearings large enough for their work, both the power spent in friction and the wear are practically negligible quantities.

Taking the axle bearings of a car, however, there are certain difficulties in arranging this method of lubrication, though it would certainly be very effective if carried out. With fairly good lubrication by ordinary means, it has been found that the friction of a journal should not exceed $\frac{1}{100}$ of the load. Suppose we take a car of a ton weight travelling at a mean speed of 20 miles an hour, with wheels 34 inches in diameter and bearings $1\frac{1}{4}$ inches in diameter, the surface speeds of the bearings will then be $\frac{1}{28}$ of the speed of the car, and the resistance per ton caused by axle friction will be $\frac{2,240}{100 \times 28} = \cdot 8$ lb. per ton. The power absorbed by this will be ·045 horse-power. If all the friction of the wheels on their axles was got rid of, the saving would be too small to measure.

In addition to this, in a car there are the bearings of the transmission gear, which will be dealt with later.

The approximate accuracy of this calculation is shown by the small resistance of a train on the railway. The resistance of any vehicle falls, roughly speaking, into four heads—The resistance of the wheel rolling on the ground, air resistance, resistance due to gradients, and axle friction. Diminishing the three first has no effect on the last, and the difference in a railway and road consists simply in diminishing the first three. Now, to pull a train on a railway, even at the very high speed of 60 miles an hour, only requires about 2 horse-power per ton, of which the greater part is spent in overcoming the rolling resistance between the wheel and the rail. It is obvious, therefore, that even at this high speed the axle friction represents less than 1 horse-power per ton, and at one-third of this speed cannot exceed $\frac{1}{4}$ horse-power.

The calculation of horse-power from the assumed coefficient of friction depends for its accuracy entirely on the lubrication being such that this is correct. It is possible, therefore, that it may exceed that assumed in the calculation. There seems, however, to be ample evidence from railway and other similar practice that the friction with any lubrication which will keep the bearings cool should not exceed at the outside $\frac{1}{2}$ horse-power per ton, including the gear-box bearings, and this would be such a small proportion of the total power, which may often up-hill be 20 to 30 horse-power, as to be imperceptible.

What, then, brought the ball bearing so extensively into use on the bicycle?

Mainly, that it would run without lubrication, while the plain one would not. Comparatively few of the modern cyclists have ever ridden on bicycles with plain bearings, or have any idea of the trouble required to keep them properly oiled. It was, in fact, quite impossible to oil them as an engine bearing is oiled, as this requires a continuous supply of oil to the bearing. Had it been possible to have a lubricator holding, say, a quarter of a pint of oil to each bearing, there is little doubt that a bicycle would have run every bit as well with plain bearings as balls; and, in fact, experience indicated that there was little difference between the two *for the first few minutes* after the plain bearing was well oiled. After that it began to get more or less dry,

and then the friction increased considerably. A still greater objection to the plain bearing was the fact that if it was not frequently oiled it became hot. Modern cyclists have hardly heard of a hot bearing on a bicycle, but they were not uncommon with plain bearings twenty years ago, and to avoid them the rider had to dismount every ten miles or so and go carefully over each bearing and give it plenty of oil, which soon came out of the bearing and spread itself over the clothes. Another point which probably helped the ball bearing a good deal is that it seems to be the cheapest to put into a cycle.

That the plain bearing is capable of very good work is proved by the fact of its staying in use in most cases where economy is of importance, and the costs of running are sufficiently accurately kept to see what is most economical. In railway work, for instance, as shown above, the axle friction is a far greater part of the total resistance of the train than it is in a car or a bicycle, and yet plain bearings are pretty well universal, though it is possible ball bearings may come in some day. There is no doubt that any way of increasing the weight-pulling power of an engine would be welcomed heartily by railway men, provided, of course, that it was reliable and the material durable.

The conclusion, then, is that *as long as the bearings can be properly lubricated*, the saving in petrol by having a ball bearing is very small. Assuming that the friction of the ball bearing is only half that of the plain one (which has not been proved), the saving is less than ¼ horse-power at an average speed of 20 miles. This would be about 8 gallons of petrol on 5,000 miles running, which may be considered to be the yearly average of a pleasure car. Against any saving there may be from the less friction must be set—(1) greater first cost; and (2) greater weight, which may add as much to the resistance of the car, especially uphill, as the friction saved, and which will, in any case, add to the expense of tyres, often the most important item in upkeep. In commercial work the saving of petrol would be very much greater, as the vehicles are run much more continuously, but in practice ball bearings are used far less in commercial work than pleasure cars.

At present the comparative results obtained in the various hill climbs, &c., in which cars with plain and ball bearings have competed, afford the only guides. Judging from these the difference in friction is negligible; as, for instance, in the thousand-mile trial of 1903 the car which showed the best horse-power at the road wheels in proportion to its cylinder capacity, had plain bearings. So had the one at the Sunrising Hill Climb in 1904. The car which went up the hills in the Scottish Reliability Trials of 1905 in the shortest time also had plain bearings, and beat several cars with larger cylinders and ball bearings.

The results with commercial vehicles will be much more instructive, as these run far more continuously than pleasure cars, and the saving in working expenses is of prime importance.

Durability.—Coming now to the question of the reliability and durability of the bearings, a matter far more important than the economy of friction, it is obvious from experience on all sorts of machines that the plain bearing will run very well and with very little attention *as long as it is big enough for its work, and is well lubricated.* There are instances of plain bearings having run many years without any adjustment, and with no perceptible wear. This proves that there is very little friction, as if there was friction there would also be wear. There are instances of bearings running every ordinary working day for ten years without adjustment where the lubrication has been good. In fact, there could be no better proof that

ball bearings are not superior to plain ones for *every* purpose than the fact that cycle makers and others, who themselves make ball bearings, have always many plain bearings in their own factories. Each case must, therefore, be taken on its merits.

The whole question, therefore, really turns on lubrication. If this is satisfactory there is probably little gain in the ball bearing. If this cannot be arranged, however, the ball bearing has a great advantage.

Taking the parts of a car in detail, the road wheels may be dealt with first. Here it is not easy to arrange either for an oil bath or for continuous lubrication. If the latter is tried, small pipes for the oil are needed all over the car. These pipes, moreover, must bend as the car rises and falls on the springs. Consequently, they are so liable to be broken by the vibration that they are not used. Road wheel bearings are generally lubricated with grease. This works very well if properly attended to, but is not very convenient, and is apt to be forgotten. Hence there is certainly an advantage in having road wheel bearings which do not want looking after.

With regard to the gear-box bearings, the case is not so clear. There is no difficulty in lubricating these, as they are all generally lubricated by a grease pot on the dashboard with pipes to all the bearings, and this works very well. Another important point here is that the gears in a gear box are in use during only a very small proportion of the time the car is running, in a modern car. That is to say, when the car is fitted with a direct drive, as most modern cars are, the gears in the gear box do no work while the car is running on its top speed, and this is the greater part of its running time. The gears are running idle, it is true, but in this case there is very little pressure on the bearings, and, consequently, the amount of friction is negligible, so that a reduction of friction is an advantage only with the lower speeds which are used for a relatively short time.

By far the most important point with regard to gear-box bearings, and one which outweighs a slight saving of petrol or a trifling increase in speed, is quietness. It is doubtful whether ball bearings are as quiet as plain bearings. The latter support a shaft along a considerable part of its length, and yet support it on a film of lubricant which has a tendency to deaden vibration. The ball bearing, on the other hand, supports it on a line (or if there are two rows of balls, two lines), but with hard metallic contact. If the pressure is very constant, as in the case of a bicycle or the road wheels of a car, this is not an objection, but with a vibratory pressure, like that from the teeth of wheels, the case is different. The matter is purely one of actual experience, and time alone can show which is the best. The ball bearing has the advantage that there is practically no wear at all in it, and, therefore, unless it fails altogether, it will not deteriorate with age; on the other hand, there should be no difficulty in making plain bearings in which the wear is also negligible. At present, as far as my experience goes, the plain bearing here has a distinct advantage, the cars with the quietest running gears having plain bearings, though it must also be admitted that there are cars with plain bearings which make plenty of noise; but the fairest comparison seems to be to take the best results in each case.

The back gear box of a live back axle, like the road wheels, is more difficult to lubricate than the change-speed gear box. Further, bevel gear generally runs more quietly than the gear-box gear.

In the case of the engine there seems no advantage whatever in the ball bearing, and it is very inconvenient to apply. As the stress on the engine

bearings are alternating there is no difficulty in lubricating the bearings thoroughly, while even if ball bearings are used, some lubrication for the engine is still necessary. An alternating stress allows of the lubricant getting much more thoroughly into the bearing than one that is constant, and, therefore, with good arrangements the surfaces never touch. Further, the alternating stresses are much more likely to injure the ball bearing than the steady stresses, and the want of an oil cushion to take the shock makes it more difficult to make the engine quiet.

For the thrust bearings of the clutch, ball bearings are thoroughly satisfactory and far more convenient than the plain bearings, as the latter are not easy to oil and soon get dry.

The objections to ball bearings are:—

1. They are slightly more costly.
2. If damaged, they are more completely broken down than a plain one.
3. They are heavier.
4. If they require renewal, they are obtainable from the makers only; but some persons claim this to be an advantage because it avoids the risk of having any bad bearings.

Reliability.—Definite and reliable experiments are wanting for settling this point, and may soon be available owing to the rapidly increasing use and number of motor buses, as in these the costs will be pretty accurately kept.

The undeservedly bad reputation of ball bearings is mainly due to some having been badly applied, and in this respect the influence of the cycle has been adverse. The usual cycle type of ball bearing is shown in fig. 294.

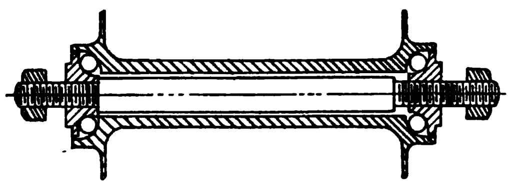


Fig. 294.

This bearing runs between cone ball races which can be adjusted. The thrust and journal loads are both taken by the same row of balls, and practically the bearings are always made in pairs, each shaft being carried on two rows of balls, each of which takes part of the journal load, and each of which takes the thrust one way. This type works well on a bicycle, in which the load is light, and there is no vibration. Further, there is no objection to running a bearing with a little slack in it. Another point is that the results of a breakage of a ball in a bicycle are not nearly so serious as in a car or other machinery.

In a bicycle, which can be led or carried, this seldom means anything worse than a compulsory walk to the nearest station; but for a fast car it probably means a serious accident, and a complete stoppage. Hence the car type of ball bearing must have a higher standard of reliability than the bicycle type.

The first point in which the modern ball bearing of best design, as used in cars and other machinery, differs from the cycle bearing is in having different rows of balls to carry the thrust and journal load. This appears to be absolutely necessary in order to get good results. The next is in abandoning the adjustable type. The reason of this appears to be, partly, that it is

not possible to arrange for adjustment in a journal bearing if the two points of contact lie in a straight line at right angles to the shaft.

Probably the chief source of failure in ball bearings is using balls and races of too small a size for the load they have to carry, as, under such circumstance, the balls and races soon wear, and then the bearing gives more trouble than a plain bearing. Ball bearings generally are not worn away like plain ones by rubbing away of the surfaces, but if they are overloaded they wear by the flaking off of the surface if of proper hardness, or rolling out if too soft. As soon as the slightest flake shows on the surface of either the ball or the race the bearing should be condemned, as the flaking will spread very rapidly and the bearing be destroyed.

Bearings are now, therefore, generally of the unadjustable type in which the balls run between ball races of definite size, which, once made, are never adjusted; but, if worn, are replaced. In making these it is necessary that the work should be of the very best or the result will be far from satis-

Fig. 295.

Fig. 296.

factory. The material must be the best, hardened and treated in the very best way, so as to leave a surface which is dead hard, and yet not brittle enough to flake off. The balls must be absolutely true to gauge and also absolutely round, as, if not, the weight will be unequally distributed on them. The bearing when put together should have practically no slack.

It is not within the scope of this work to go into the detail of the manufacture of the bearings or the relative merits of the smaller variations of arrangement adopted by different makers. Ball bearings are at present specialities, and it is not likely that motor car makers will make them for themselves except when they are working on the very largest scale, and it is a rather doubtful matter even then. In this work, therefore, the arrangements adopted by one maker of high standing will alone be described. Those who wish to go further into the matter are referred to different makers' catalogues.

By the courtesy of Messrs. The Hoffmann Manufacturing Co., Ltd., I am able to give some particulars of the bearings made by them, and also of the proper applications of them to motor cars.

The Hoffmann bearing for taking a load on a shaft is shown in fig. 295. It consists of two concentric rings with shallow grooves in them, and a row of balls between, separated by a cage. The rings are quite continuous, and, therefore, when the balls are evenly spaced between them, the two rings and the balls cannot be taken apart. In order to put them together or take them apart the balls are all shifted round to one side, when the inner ring can be moved out of centre with the outer, and the balls inserted or removed.

In the Hoffmann bearing a cage is provided to keep the balls at even distances, this cage being made of bronze, and put on the balls in halves, which are rivetted together. This bearing is only intended to take a journal load, and not to take any end load on the shaft at all, and it is most im-

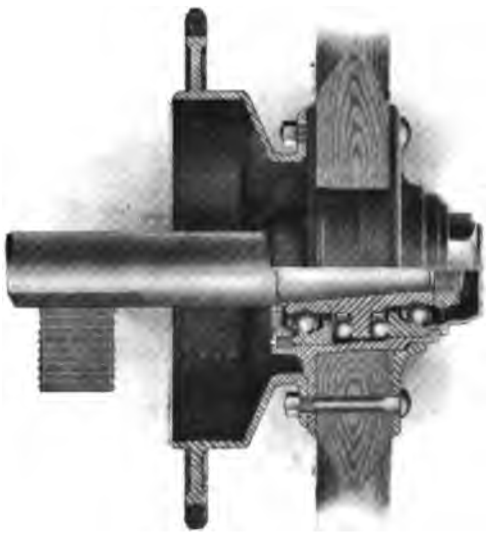


Fig. 297.

portant that it should not have any end stress put on it, as the result would be that the curves of the grooves would act as most powerful wedges, and soon seriously damage both the balls and the ball races.

A special type of bearing, shown in fig. 296, is made for taking end thrusts, the same principle being used as in the last, which is to make the line of contact of the balls coincide with the direction of the stress on the bearing. This bearing will, therefore, only take end load, and would be distorted if subjected to a radial load. It will be understood that, though the bearings constructed to carry journal load are not suitable for carrying any end thrust, yet the grooves are sufficiently deep to retain a shaft in place endways if there is no special end thrust upon it. Thus in a small gear box, if the shafts are so arranged that there is no thrust on them from the clutch or any other part of the gear, the ordinary bearing shown in fig. 295 will be found satisfactory.

On the other hand, for such parts as the bearings of the road wheels, a bearing which provides for the taking of the end thrust is necessary, and as this has to be taken in both directions, it is necessary that there shall be four complete rows of balls. Such a bearing for a wheel running on a dead axle is shown in fig. 297. In this there are the two rows of balls, one at each end, to carry the weight of the car, and also a row to take the end thrust in each direction. The stress produced by side pressure on the tyre of the wheel, as when going round corners, is resisted by the two centre rows of balls. This hub is suitable for the bearings of either the back wheels of chain-driven cars, or the front wheels of any car. The hubs are made in a variety of sizes to suit all cars, as in the table below. The

SIZE OF BEARINGS.	1⅛	1¼	1 9/16	1¾	2	2¼	2½
Diam. of large journal balls, .	½	⅝	⅝	¾	⅞	1	1⅛
,, small ,, .	7/16	½	9/16	⅝	¾	⅞	1
,, thrust balls (2 rows), .	7/16	½	9/16	⅝	¾	⅞	1
Number of balls in each row, .	10	10	10	10	10	10	10
Diam. of flange, . . . in.	6¼	7¼	7⅝	8½	9⅞	11	13
,, axle, small end, . in.	¾	⅞	1 1/16	1 5/32	1 5/16	1½	1⅝
,, ,, large end, . in.	1⅛	1⅜	1 9/16	1¾	2	2¼	2½
,, ,, body, . . in.	1½	1¾	1⅞	2	2½	2½	2½
,, end of housing, . in.	3½	4¼	4 7/16	5	6	6¾	7½
,, body ,, . in.	3¼	3¾	4⅛	4 9/16	5⅜	6¼	7
,, taper ,, . in.	3 7/16	4⅜	4⅜	5⅛	5⅞	6⅛	7¼
Degrees of taper on shaft, . .	4°	4°	4°	4°	4°	4°	4°
End of cone to end of housing, in.	11/32	13/32	13/32	7/16	½	9/16	9/16
Length of taper inside cone, in.	2 11/16	3 47/64	3 47/64	4¼	4 53/64	5 41/64	6¼
,, straight part inside cone, . . in.	33/64	7/64	33/64	13/64	9/64	27/64	7/16
End of cone to inside of grease cap, ... in.	1 1/64	1 9/32	1 5/32	1 15/64	1 15/32	1⅝	1⅜
Undercut width between flanges, in.	1 11/32	1⅝	1⅝	1⅝	1⅞	2⅛	2⅛
Thickness of flange, . . in.	¼	¼	¼	5/16	5/16	5/16	½
Inside edge of flange to edge of taper, ... in.	27/32	31/32	1	1	1 5/16	1⅜	1½
Inside flange over end of housing, in.	1 11/64	1⅜	1 7/16	1 43/64	1 29/32	2¼	2½
Width between flanges, . in.	1¼	1½	1½	1½	1¾	2	2
Overall length, . . . in.	4¾	5⅜	5¾	6⅝	7¼	8⅛	8⅞

fixed sleeve of these bearings is ground to a taper for the reception of the end of the axle, on to which it is fastened by a screw and nut. It can easily be removed by unscrewing this, and to facilitate removal a service cap is supplied, fitting the same thread as the cap on the wheel, which will pull the hub off the axle.

Fig. 298 shows this type of hub as applied to a steering wheel, and also shows the application of a ball-thrust collar of the steering pivot. Whether the latter is worth applying when a worm and wheel steering is used to make the steering steadier by increasing the friction seems very doubtful.

Applications.—With an axle of the live type several arrangements may be used. We have here to provide for the weight of the car, the end pressure due to the side stress of the wheels on the ground, and the thrust of the bevel gear. As it is not advisable to provide thrust bearings

at both ends of a shaft, the thrust should be confined either to the gear box or to the end of the axle. In either case the end which has not got the

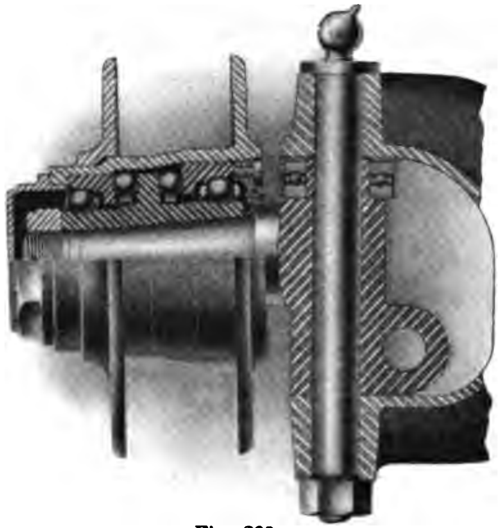


Fig. 298.

thrust combined with it will have a bearing of the type of fig. 295 to carry the weight.

The former of the two arrangements is shown in fig. 299. This is the simpler of the two, as there is only one double-thrust bearing on one side of

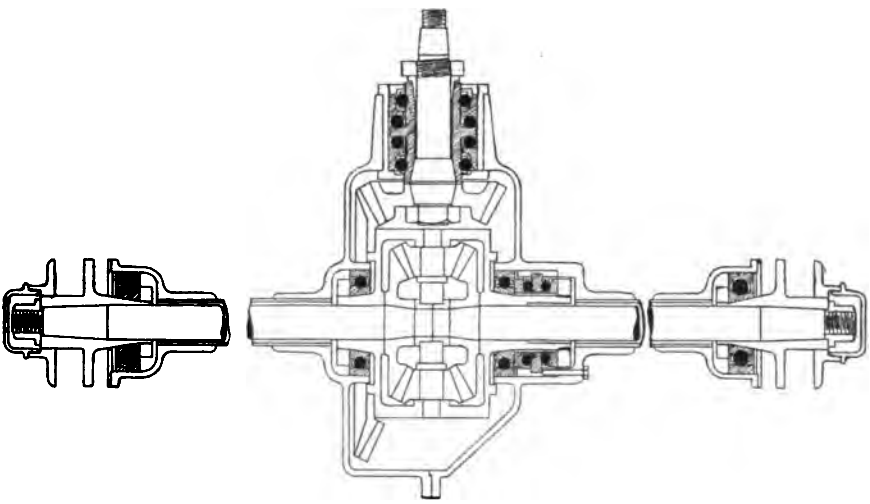


Fig. 299.

the differential, and two rows of balls to take the weight. The differential and cage, &c., in this arrangement will be so arranged that there can be no

end movement between the ends of the shafts, so that the side thrust which comes on the outside wheel in going round a corner will be transmitted through the axle to the opposite axle, and taken on the thrust bearing opposing it. The bearings which carry the weight, on the other hand, will be of

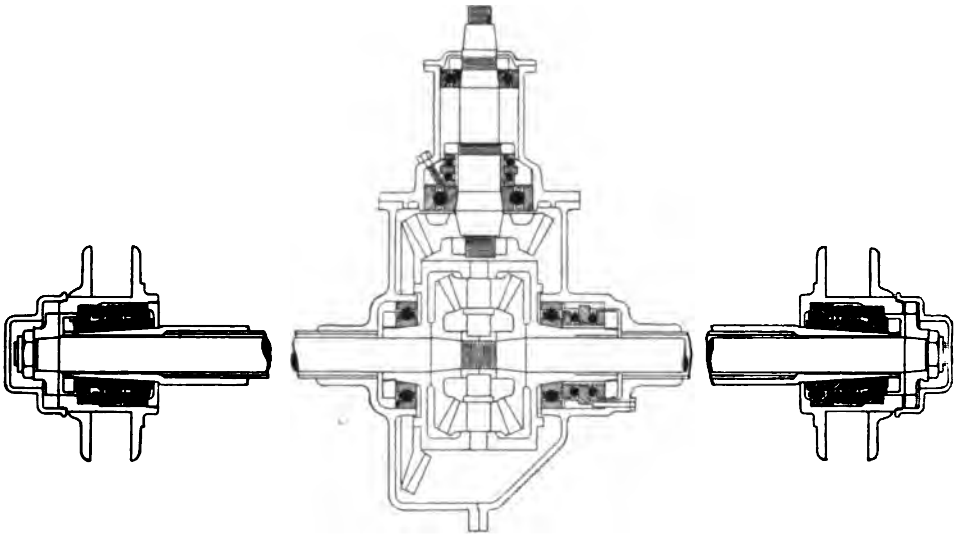


Fig. 300.

the type of fig. 295, fixed on the shaft and simply slipped into the housing, so that it will be quite impossible for any end thrust to come on them.

Fig. 300 shows the arrangement of ball bearings in an axle of the type in which the load is carried on the axle tube, for which see Chapter xiv., with a different arrangement of bearing on the pinion shaft.

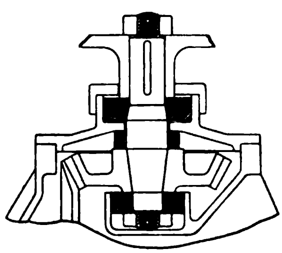


Fig. 301.

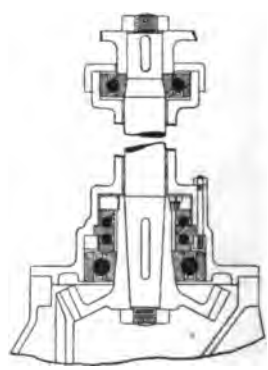


Fig. 302.

The pinion driving the bevel gear may be carried on a single bearing, overhung, as in fig. 299, or carried with a bearing beyond the pinion, as in fig. 301. The former has many advantages in the general design of the gear

box, as, in the case of a bearing beyond the pinion, there has to be provision for getting it in, and also it is not always easy to make a good arrangement of differential, &c., without the bearing fouling. Messrs. Hoffmann also inform me that they prefer the overhung arrangement, as shown in fig. 299, as the whole bearing is then complete in itself, and is of the same type as that shown in fig. 297, in which there are four rows of balls for taking the stress off the side thrust of the pinion.

If the propeller shaft is carried in a long tube, as in fig. 215, the bearing at the forward end should be of the single-row type, and there is no necessity for a bearing at the gear box end of the four-row type, though this can be used quite well if desired. It is, however, generally more convenient to have a single-row bearing and a thrust bearing, as in fig. 302. The thrust from the pinion is always in the same direction in theory, but in practice it is

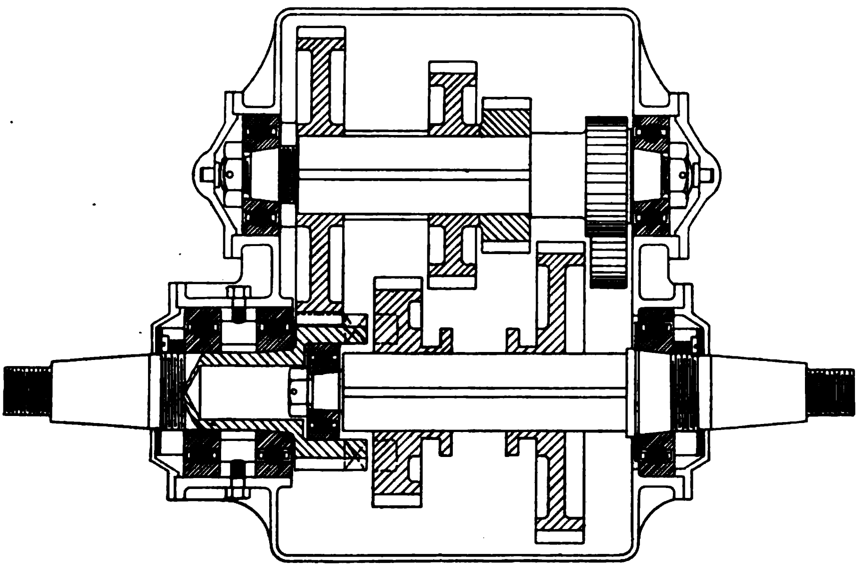


Fig. 303.

found necessary to hold all shafts, upon which gear wheels are mounted, firmly in each direction by means of a double-ball thrust bearing. This ensures the gear wheels keeping in proper pitch, and reduces both the wear upon them and the noise.

Occasionally a ball bearing is placed in the differential between the differential gear wheels and the cage of the differential to take the thrust of the differential gears. Sometimes one is put between the two ends of the axle to take the thrust from the one to the other in going round a corner. Neither of these seems necessary, as the motion between them is simply the difference of speed in the two wheels in going round a corner, so that a good plain bearing with plenty of hard wear-resisting surface is all that is required.

In applying ball bearings to a gear box there is no end thrust to provide for other than what there may be from the clutch, but this is generally otherwise provided for. Consequently, the shafts in the gear box may be

carried on simple bearings of the type of fig. 295. The exact arrangement will depend largely on the general arrangement of the gear box. If the shafts are carried in bearings held down by caps, as in fig. 241, the rings of the ball bearing will be held down in exactly the same way as the bushes of the plain bearings. The shafts must, in all cases, be so arranged that the rings of the bearing can be slipped on, as split ball bearings cannot be employed. Otherwise, no special alteration in the general design is required.

If the gear box is so arranged that the bushes are slipped on to the shafts, the arrangement of the parts is similar, the only difference being the substitution of ball races for bushes, which will be held firmly on their shafts. Fig. 303 shows the general arrangement of the ball races for a gear box similar to fig. 237. The front bearing necessarily has two rows of balls, as the spindle carries a gear wheel and its length is very short. Otherwise, single row bearings are sufficient.

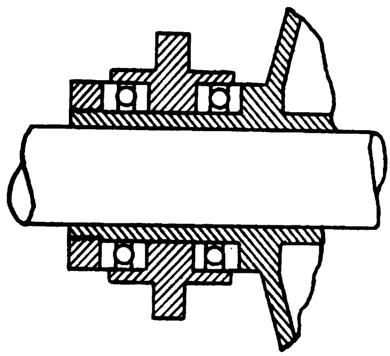


Fig. 304.

Dirt must be kept out of ball bearings, especially those with the gear box, as the dirt in this often consists of small steel chips from the gear wheels; this may be done by fitting leather washers on each end of the bearing. Messrs. Hoffmann, preferentially, use grease for bearings, and, therefore, would exclude oil from them, as this is liable to wash the grease out.

The application of ball bearings to the thrust collars of a clutch is very simple, and is shown in fig. 304.

One point that deserves attention is that the principles on which ball and plain bearings run are rather different, and, therefore, they should not be used together in such a way that the wear of the plain bearing brings more than its fair share of work on the ball bearing. In some cases, for instance, the shaft carrying the bevel pinion of a live back axle, has one ball bearing and one plain one. Ball bearings showing perceptible signs of wear are at once condemned and renewed, but plain bearings are allowed a certain latitude in the way of wear before being renewed. The result of using both may sometimes be that when the plain bearings are worn, the stress is transferred to the ball bearing. This must also be remembered in considering the application of ball thrusts with plain bearings for the end load and *vice versâ*. If the wear of the plain bearings is liable to put undue stresses on the ball bearings there will be trouble. The best plan evidently is to use one kind of bearing only in the same part of the machine.

It will be noticed that the bearings are, in all cases, of the two-point type with the load carried exactly through the opposite diameter of the balls in line with the load. This ensures a pure rolling action.

The journal bearings carry no side thrust, and the thrust bearings carry no load. In fact, the bearings are very quickly damaged if these loads come upon them. It is, therefore, essential in the case of journal bearings to see that the outer ring is a loose sliding fit in its housing. If this is done, the ring will find its correct position, with regard to the inner ring or cone.

In like manner, the thrust bearings must be held by an efficient journal

bearing, so that the two ball races are held exactly concentric with each other.

Messrs. Hoffmann consider it important that the cone or inner ball race is tightly held upon the shaft, and the best method of doing this is to mount it upon a conical seating turned directly on the shaft itself, and held on the shaft by means of a nut. This method is shown in the illustrations. Parallel holes in the cone are not recommended, unless they can be mounted immediately upon one end of the shaft, in which case they can be driven on tightly. The tendency of these cones to creep round the shaft is considerable, not due to the friction of the ball bearing, but to the inner circumference of the hole in the cone being larger than the outer circumference on the shaft on which it is mounted.

As mentioned above, ball bearings are largely the speciality of firms who make them on a large scale, and it is impossible to go into all the differences of construction favoured by the different firms. The general principles indicated above are common to all, but the details differ considerably, both in the method of putting the balls in and in the use of a separating cage. For more detailed information, both as to the construction of bearings of other makers and their application, the reader should consult the maker's catalogue.

CHAPTER XVIII.

BODIES.

Bodies.—The upholstering, &c., and construction of bodies do not come within the scope of this book, as the bodies are generally supplied by coach-builders or by a separate department in a car-builder's factory.

The simplest body has seats for two persons, and presents little variation;

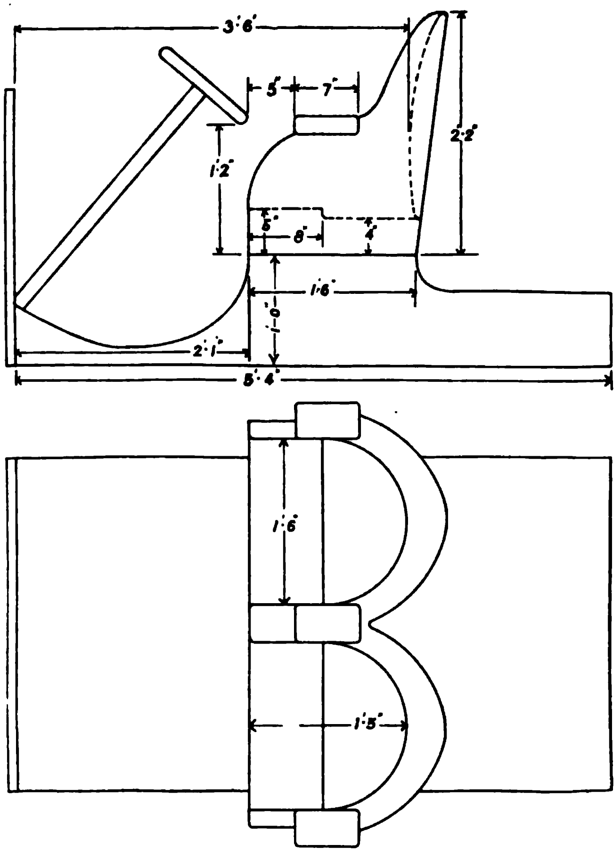


Fig. 305.

there may be either one wide seat or two separate seats with a round back to each seat, as in fig. 305; they are generally called "bucket" seats, but the reason for the appellation is not at all clear. The comfort of a car depends largely upon the roominess of the seats, which should be ample for stretching

the legs and for the thick clothes usually worn. In many cars the seats are too cramped, especially in French made cars, Frenchmen being generally smaller than Englishmen. The advantages of the bucket seats are that they afford a good seathold, even if slightly separated, are very comfortable and much warmer than other kinds. The proportions given in fig. 305 will be found to be comfortable by most persons. The floor board in front is now generally sloped up, as shown, and is very comfortable.

The dashboard should be far enough away to give ample room for the

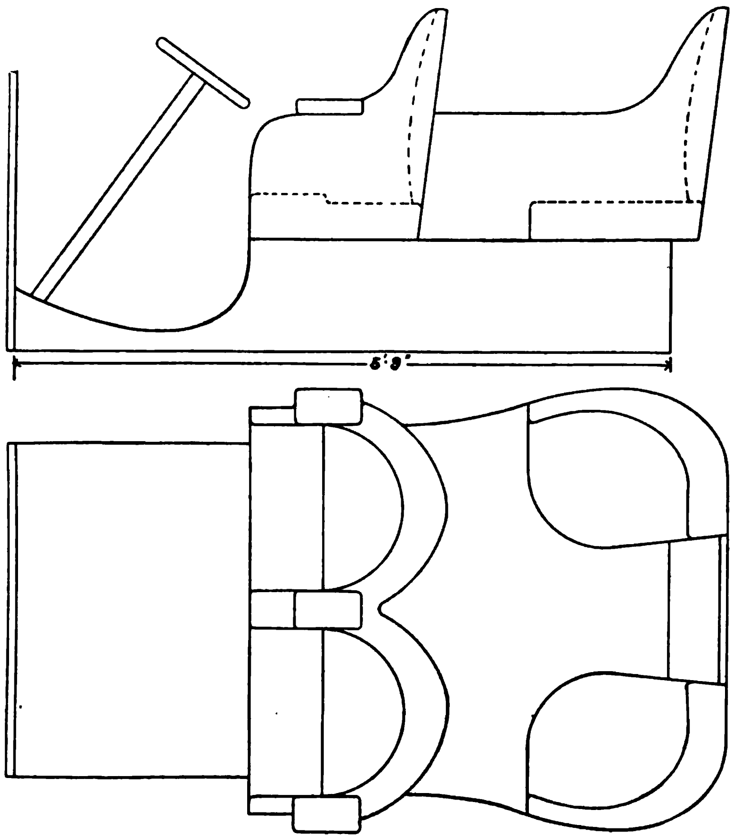


Fig. 306.

legs, and the floor be wide enough to rest the feet on, clear of the pedals. In some cars the frame is narrowed under the floor boards, which are also narrowed to the same extent, but this is very uncomfortable, while there is no good reason for narrowing either the frame or the floor boards.

Bodies intended for more than two people have more variety in their seating, but it is generally admitted that everyone must face forwards. The most compact body for this is the small tonneau (fig. 306). Although rather out of fashion just now this is a very useful body, although there is some difficulty in making back seats as comfortable as the front. Still there is

a great demand for a car which will carry two people and yet allow of an occasional lift being given to one or two others. The tonneau just fills this condition, as, owing to the shortness of its body, the car is very light for four people. For those who cannot afford to keep more than one car, which is generally needed for one person or for two persons only, a four-seated car weighing nearly a ton is out of the question; the small tonneau will satisfy their needs, especially as the back part can be made to lift off, so as to leave the platform available for luggage.

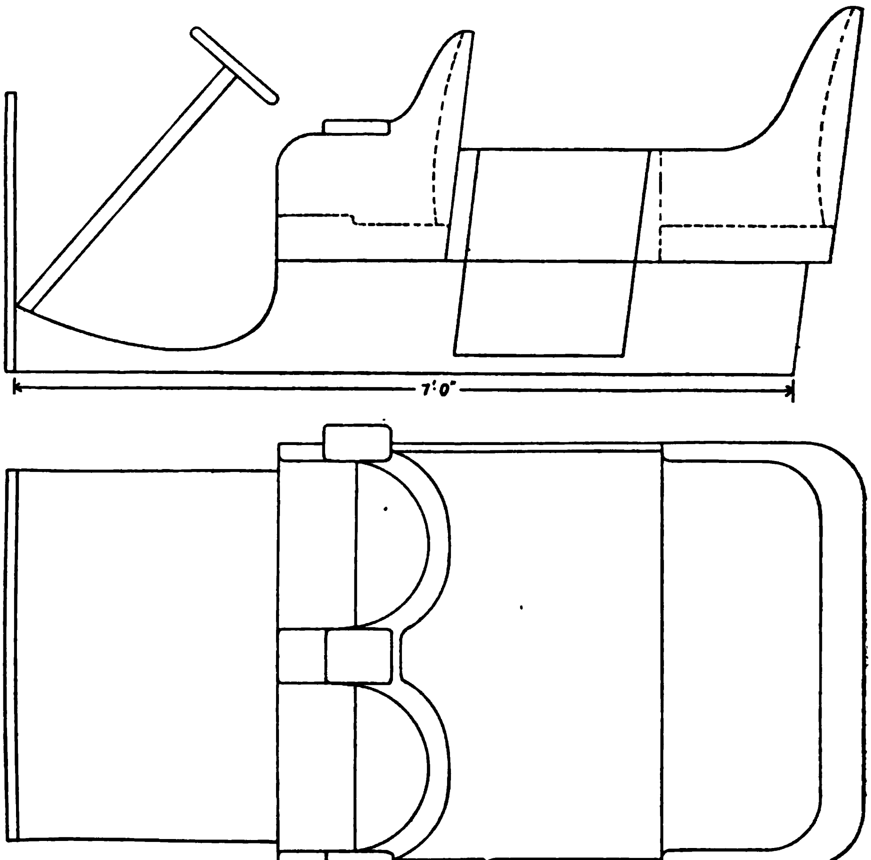


Fig. 307.

The next form of body to be noticed is the double phaeton of various designs, the entrance to which is in front. In the usual forms, either one side or the whole of the front seat is hinged. In either case the car is only a little longer and heavier than the tonneau.

In the next design the body has a side entrance (fig. 307), and is certainly the most convenient type for four people, but in order to make it a really comfortable one it needs that the car should be long in the wheel

base in order to get the door to the back seats clear of the wheels. This means a considerable increase of weight, as the frame has to be longer as well as the body, with stronger, and therefore heavier, scantlings, and also a larger engine.

Closed carriages are variations of the above. For doctors the chief demand will be for a closed carriage to carry two—that is, a two-seated car covered in. If room for two and a driver only is desired, a roof may be put over the back seat of an ordinary side-entrance car, or if a lighter body is required, the left-hand seat of the double phaeton may be dispensed with, and a roof put over the back seats. This will be much shorter, and shorter than the car with a side entrance, the disadvantage being that it can be entered and left on one side only. In many cases the back seats are covered with a roof that can be lowered.

The best body for carrying more than four persons is an undecided point. The cheapest and lightest is the waggonette. This can be made of any length to carry a large number of people, and be either open or covered, in which case it becomes a 'bus; or a detachable top may be used, in which case such a car would be most invaluable at a large country house. Its disadvantage is that it is not so pleasant to sit sideways as it is forwards. Or one may multiply the side-entrance car with any number of seats up to the 36-seated char-a-banc.

The point in the bodies which most affects the designer of the chassis is their varying length and weight. It is not quite clear how this difficulty is going to be met in the future, though at present there is a tendency to make a given chassis strong enough to carry any body that may be put on it, and to make that a standard for a given-sized engine. This means that if this chassis is used for a light body, the back axle and frame are much heavier than is necessary.

The better plan, perhaps, would be to have several patterns of back and front axles, engines, and gear boxes suitable for a definite wheel track and width of frame, and to combine these according to what is wanted. This has been carried out by one firm with very satisfactory results. Thus, for a very fast car to carry two persons a light back axle is used with a high gear, while for a heavy body the combination would be a heavy back axle, small engine, gear box, and suitably low gear. Springs would, like the axles, be standardised for a given weight. Three or four axles, frames, engines, and gear boxes would then combine into a great many different variations of car without any special patterns being needed.

As mentioned, the main point which affects the chassis design is the length of the body, and the foregoing figures will give a rough idea of the space required.

CHAPTER XIX.

FACTORS OF SAFETY—CALCULATIONS OF STRESS.

Definition.—A piece of metal when subjected to a moderate stress is stretched by it, but resumes its original length when the stress is taken off, but if the stress is increased beyond a certain point it loses the capability of contracting when the stress is removed. This point is called the "elastic limit." If the stress is increased still more the piece will break. It is obvious that the load on it, consistent with safety, should not be equal to the breaking strength, but be something below it. Thus, if the load actually put on the part is one quarter of that at which it is calculated the material will break according to the tests in a testing machine, we say we have a "factor of safety" of four.

If the load was always applied in the same way as when tested, the metal would be safe for all loads nearly up to its elastic limit. But, as the stresses it will have to bear cannot be accurately determined, and as the metal deteriorates in strength in proportion to the intensity and intermittency of the stress, the load should be well within the elastic limit.

Taking the first point, it is obvious that stresses on a part (let us say an axle) when the machine is standing will be far less than those induced by the shocks, when a car goes over rough ground, &c. This applies to all "live loads"—*i.e.*, to all moving loads—and to all parts which support such loads (*e.g.*, carriages, locomotives, railways, bridges, roads, &c.). The amount of shock may vary enormously, depending, in the case of vehicles, mainly on the nature of the road and tyre. It will naturally not be so great with pneumatic as with iron tyres. It will also, to a certain extent, vary with the speed.

Fatigue.—Deterioration is a matter of great importance in motor-car work, for a piece of metal becomes weaker with every incidence of stress, so that when this has been repeated a definite number of times, depending on its intensity and direction, the metal becomes tired out and breaks. The greatest variation there can be is a complete reversal—that is to say, the stress coming first on one side of the bar and then on the other. This takes place in a live axle as it revolves. In such a case, it is found that the load a bar will carry without breaking depends on the number of times the stress is reversed.

For full details on this subject, the special works devoted to it must be consulted; but for the present purpose the following brief outline will suffice. The most important experiments in this connection are those of Wöhler, on revolving spindles made of the materials indicated. These were loaded and then revolved until they broke; the results of some of his tests are given in fig. 308. It will be seen that the breaking load decreases more or less regularly with the number of revolutions run. The curves only give the results up to 1,600,000 revolutions, but the experiments were continued in many cases up to 150,000,000 revolutions.

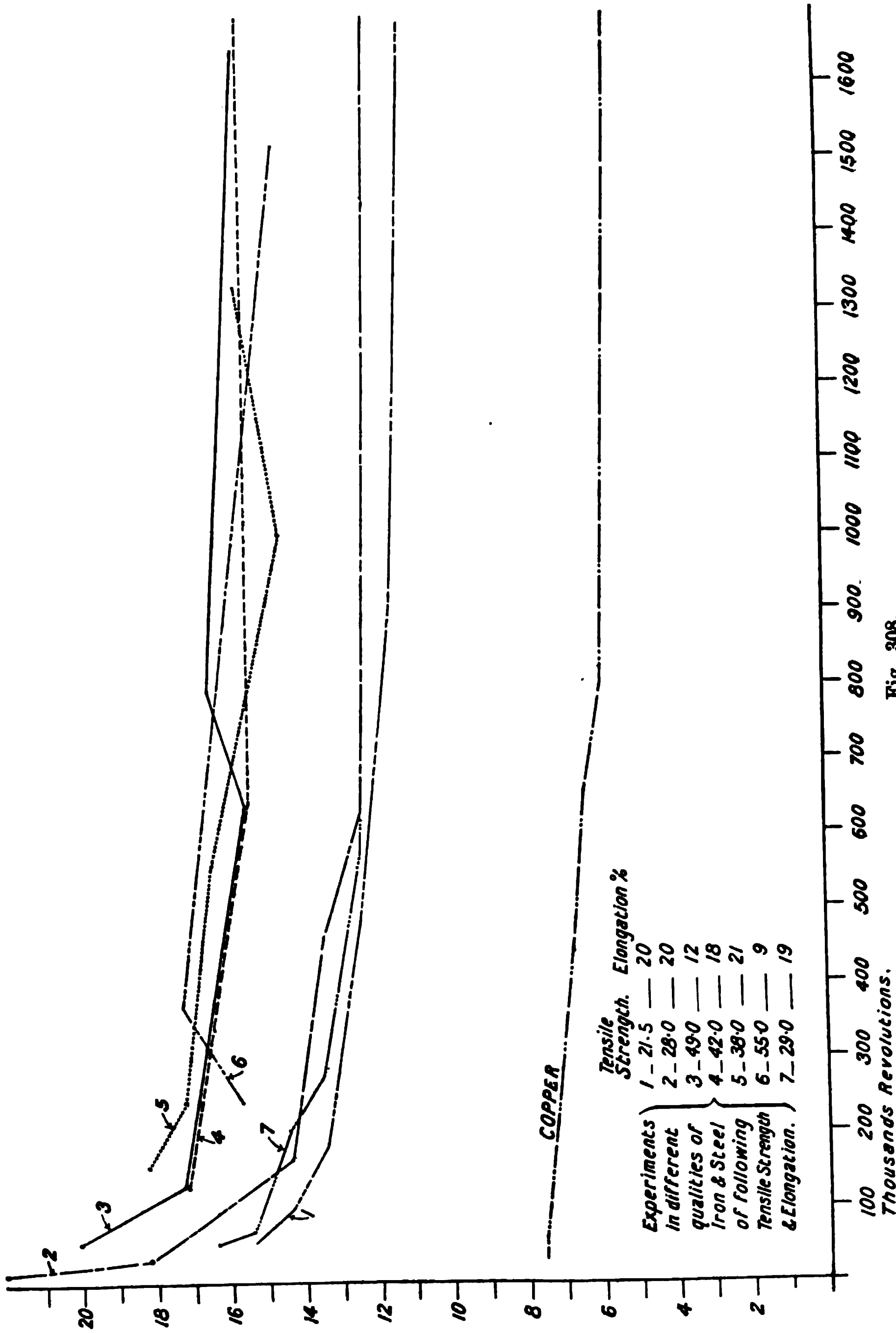

COPPER
Experiments in different qualities of Iron & Steel of following Tensile Strength & Elongation.
Tensile Strength. Elongation %
1 _ 21·5 — 20
2 _ 28·0 — 20
3 _ 49·0 — 12
4 _ 42·0 — 18
5 _ 38·0 — 21
6 _ 55·0 — 9
7 _ 29·0 — 19
Thousands Revolutions.
100 200 300 400 500 600 700 800 900 1000 1100 1200 1300 1400 1500 1600
2 4 6 8 10 12 14 16 18 20

Fig. 308.

The net result of all this is very interesting both to the motor-car builder and user, as it shows quite conclusively that the stress which may be allowed on a part depends entirely on the time it is desired it should last without breaking. For instance, a bar of iron stood a stress of over 14 tons per square inch for 99,000 revolutions before breaking, yet broke at 8·6 tons, or little more than half this, with 19,000,000 revolutions.

These observations have been confirmed by other investigators, and it is now taken as an axiom that the load any particular material will stand depends on the number of alternations of stress or variations of stress to which it is subjected. Variations of stress do not diminish the strength of a material to the extent to which alternations of stress do, but they deteriorate it a great deal. In the case of springs, fixed axles, frames, &c., the stress is always in one direction, but varies in amount every time the car goes over the slightest bump or hollow in the road—that is to say, about every foot the car travels. It is also different for every variation in the weight in the car.

A curious point which has been brought out in these experiments is that, although the material may have been deteriorated to the point where a very small number of revolutions will cause it to break, it will not show any diminution of strength in an ordinary testing machine. That is to say, it will carry as high a load before it breaks, and will also break with all the indications of toughness which a piece of the same material does which has *not* been subject to any alternation of stress.

It is well known to those who have had any engineering experience that the nature of the break in any part which breaks in practical work is usually quite different to that in a testing machine. In the latter the material will stretch materially before breaking, but in practice the mode of breakage is that of a perfectly brittle material. The breakages under alternations of stress are exactly of this kind. A further point brought out by these experiments is that there is for every material a particular load below which it will stand a practically unlimited number of variations or repetitions of stress without breakage. This is, of course, obvious in ordinary engineering, as machines run in many cases for years. Railway axles often run as many as 300,000,000 revolutions before being withdrawn from service, yet with hundreds of thousands of them at work a broken one is hardly known.

Pleasure cars are on rather a different footing. The ordinary pleasure car runs perhaps 5,000 miles a year, and in three or four years often begins to give great trouble from breakages, say, in 20,000 miles' running. Now, a revolving axle with a 32-inch wheel makes about 500 revolutions a mile, so this represents about 10,000,000 revolutions only. This leads to two very important conclusions, which have affected the whole of the motor industry.

1. There is no such thing as an absolute breaking stress up to which a material is safe. The load which may be put on it is simply dependent on the time it is desired that the part shall last, and the percentage of breakages which may be tolerated.

2. That it is perfectly possible to have no breakages at all, and to have parts carry their load for a practically unlimited time if the material is suitable and the load is not too great.

The first point is one which should be thoroughly appreciated by motor users. A machine may be made with the necessary factor of safety for running continuously for years without breaking. If similar machines are made with lighter parts, it is not found that there is any point up to which they stand perfectly, and then suddenly break down; the experience is that

after a certain time there is a certain percentage of breakage. Suppose any machine be taken, such as a locomotive, in which the parts are made strong enough to run permanently, but not heavier than necessary. If the parts are lightened, say 10 per cent., it is quite likely the engines might run for two or three years, say 250,000 miles, without any breakages occurring. After this something would break from time to time. If made still lighter, the percentage of breakages would be higher, and these would occur sooner.

Now, speed has been one of the chief recommendatory features in the sale of cars, consequently the parts have been made lighter relatively to the load in order to secure greater speed; and the question for the public to consider is how far speed has been gained at the cost of reliability, and, in judging the merits of a new car, they should bear in mind the percentage of breakages in cars of the same type which have been in use for a number of years.

Effects of Shock.—In the experiments quoted above the stresses were all simple alternating ones, and there was no shock. Further, the test pieces were all made with good round corners, so that there were no great local stresses. In practice the conditions of running are very different. There is always a considerable amount of shock and vibration in running machinery, and it is impossible to so design it that there are no sharp corners. The effect of both these is to considerably reduce the strength of the parts, and therefore the load they will carry, particularly with a varying stress.

These considerations will show how the present factors of safety in various machines have been arrived at. They are simply the experience of what will and what will not run for a long time without breaking. No engineer desires to put in parts larger, heavier, and more expensive than are needed, but if it is found that parts loaded with more than a certain stress are liable to break, these parts are made stronger in the new machines so as to avoid breakages in them. If general users of machines had desired lightness as much as many of the users of motors have done, and were willing to put up with the amount of repairs which the latter are liable to require, engineers would have made lighter machines than are now used. The usual way of expressing the load on a part is to give the stress per square inch of the section. Dividing the breaking stress of the material, as given in a testing machine, by this gives us the factor of safety. The stresses vary a good deal in different classes of work according to the amount of vibration, &c., that the parts are subject to, and also according to the use to which machines are put. The greater the importance of reliability the lower the stress permitted. The stresses usual in such machines as stationary engines, &c., which are not worked under the same conditions as motor car engines, are of little interest here. Marine, locomotive, and such like engines, on the other hand, are subject to very similar conditions. Weight in these is of the greatest importance, provided it can be reduced without sacrificing reliability. In these the calculated stresses allowed on parts which have the stress reversed is generally below 10,000 lbs. per square inch; sometimes it is 12,000, but seldom beyond this, except for special work where lightness is everything and cost of repairs of little importance.

The above stresses are for ordinary mild steel of the best quality, which will have a breaking stress of about 60,000 lbs. to 70,000 lbs per square inch. This will give a factor of safety of about 6 for ordinary work accompanied by much shock.

In considering the stresses to be allowed on various parts of cars regard must be had to the relative duration of the stresses under full load and

under partial load. In a locomotive axle, for instance, the full stress is on it all the time the locomotive is running; as far as the weight on the parts of a car are concerned we ought to allow for this too, although it is very common for cars constructed to carry four people to be generally used for carrying two. The maximum driving stresses on such parts as live axles are only felt by them when the brake is put on suddenly or when the car is going up a very steep hill, that is for a very small part of the running time. Judging from other machines a fair allowance would be about 12,000 lbs. for the parts subject to reversal of stress, and 15,000 lbs. for those which are not, such as fixed axles, frames, &c.

Materials.—The question of the strength of the material affects the question of the load which may be safely allowed, and in many cases the steel used for cars is not of the same quality as that used for other purposes. At first sight it would seem that the stronger the steel the higher would be the load that could safely be put on it, and that this would be in the same proportion—*i.e.*, that the same factor of safety would always be equally safe—but, as will be shown, this is not the case. In a tensile test the data are—the point at which the steel begins to stretch permanently, the total amount it stretches before it breaks, and the amount to which it contracts at the moment it breaks. Fig. 309 shows roughly the shape of

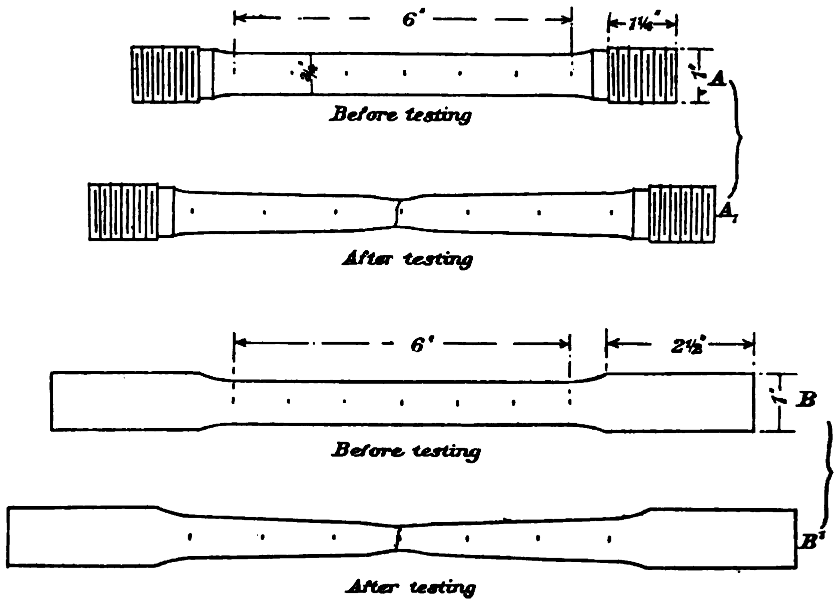


Fig. 309.

the break, from which it will be seen that when tested in this way a good piece of steel stretches a great deal before it breaks, and that most of the stretch is near the place where it breaks. The expressions for these results are—1, elastic limit; 2, elongation per cent.; 3, contraction of area per cent.; and 4, breaking stress.

The best qualities of wrought iron will have the following characteristics,

approximately:—Breaking stress, 25 tons per square inch; elastic limit, 16 tons; elongation, 15 per cent. on 8 inches; contraction of area, 20 per cent.

As the elongation is always mainly near the point of fracture the length of the test piece should always be stated.

Steel.—Steel is a name now given to all compounds of iron, carbon, &c., though originally it was used for iron which had sufficient carbon to allow of it being hardened and tempered. Mild steel, as generally used in engineering, has a tensile strength of about 30 tons, an elastic limit of 17 tons, an elongation of 20 to 25 per cent in 8 inches, and contraction of area equal to 45 to 50 per cent. This is the material generally used by engineers for the working parts of their engines, &c., as they consider it gives the best average combination of strength and reliability.

Steels of very much higher strengths have been known for many years, and have been used for special purposes. Steel of the kind specified above is iron with a very small proportion of carbon and traces of other elements in it; the strength can be considerably modified by varying the proportion of carbon. By reducing the amount of carbon the steel becomes more ductile, but not so strong—*i.e.*, the elongation is over 30 per cent. on 8 inches, with a tensile strength of about 22 to 25 tons; and by adding carbon the ductility is lessened and the tensile strength increased. Steels of over 70 tons tensile strength with low elongation have been thus made, and, by hardening, the strength has been raised to nearly or quite 100 tons, but in this case there has been no elongation at all, the steel being quite brittle. It is not surprising that in practice, where they are subject to shock, the power of steels to carry loads depends more on their toughness or ductility than their tensile strength; hence steels with a high tensile strength but low elongation not only will not carry loads proportionate to their strength, but will not even carry as great loads as steels of lower strength but greater elongation. What proportion of strength to elongation gives the best results in any particular place is a matter of experience, but for general engineering use the above is fairly general. Railway engineers, however, often use harder steel with a good deal higher tensile strength than 30 tons.

Of late years a great many special steels have been brought out in which high tensile strength and high elongation have been combined in a most remarkable degree. It might be thought that, in this case, we were quite safe in using such material, and increasing the load in proportion to its strength, as there was ample evidence of its toughness in its high elongation and contraction of area. There is every likelihood of many of the new steels being a considerable improvement on the old, but it hardly seems safe at present to assume that their strength on a load which alternates is in direct proportion to their strength and elongation in a testing machine. The influence of repeated stress and vibration is very different from that of a steady load in a testing machine, and it is well known that there are materials which give good results in the latter which do not give good results in practice. In fact, although, as has been seen above, ordinary mild steel has both higher tensile strength and higher elongation and contraction of area than wrought iron, it is quite well known that there are places where the latter will stand a heavier load than the former without breaking. In some cases even copper will stand more than either, although the results obtained with it when tested are very low. In deciding as to the use of a steel the results

given by a long experience under actual working conditions are far more reliable than those afforded with a testing machine. Many special alloys, both of steel and other metals, are such that they cannot be made absolutely uniform in shop practice, although they can be in laboratory experiments. Some also, though giving excellent results for a short time, are liable to develop cracks under alternating stress and shock, which spreading through the mass cause it to break with a load less than ordinary mild steel would stand safely.

Of late there has been a tendency to consider that a shock test is in many ways a better guide to the load a material will carry than any form of static test. In a shock test the test piece is broken either by a succession of blows, in which case the number required to break it is taken; or by one blow, in which case the energy absorbed in breaking it is taken. These tests have given remarkable results, and have shown that in many cases materials which give excellent results both in toughness and strength on a static test break easily under shock. Further, it has been shown in some cases where parts apparently of good material have broken, that under shock test the material was brittle, although tough enough under static test. This test shows that many steels, which give most wonderful results in combination of tensile strength and elongation, do not give any better results than ordinary mild steel under shock test.

For the greater part of automobile work there is probably little doubt that steel with a tensile strength of about 35 to 40 tons will stand higher stresses than the milder forms. Such steel is, in fact, being used in lorries for axles to carry loads under which milder steel has broken. With regard to the stronger steels of special composition, the case is more doubtful. In fact, in pleasure cars the actual stresses, as far as I have been able to calculate them, are seldom higher than ordinary mild steel will stand perfectly. Very high tension steel is useless if it will not safely carry heavier loads than ordinary mild steel.

The following instance shows how delusive short tests are. An ordinary mild steel when tested indicated a tensile strength of 35·6 tons per square inch, and an elongation on 2 inches of 31 per cent. An axle made from it was used in a car which had a maximum combined twisting and bending stress, when the car was loaded with four people, of about 21,000 lbs. per square inch, according to the method of calculation given in the next chapter. The axle was of the ordinary live type, carrying the load on the revolving part, and was $1\frac{3}{8}$ inches in diameter. This axle ran well for more than 10,000 miles over the roughest roads, and often carried considerably more weight than the four people the car was seated for. In fact, the car once carried eleven people, which would bring the total weight of car and load up to about 43 cwts., or nearly twice what would be reckoned on in the calculation of stress; under such conditions the axle must have been running under some 30,000 lbs. calculated stress. Nothing could seem to be more satisfactory, and had this been of a special steel it might have been held as conclusive proof that it would stand far higher stresses than ordinary mild steel would do. Yet this axle subsequently broke before it had run 15,000 miles when the car was running with a very moderate load on a nearly flat road; consequently there was no special stress on it. Clearly, it broke through the fatigue of alternating stress and shock.

When considering special steels the important point to know is if they will *permanently* retain their superiority over ordinary steel. The above

instance may serve as a rough standard of what ordinary steel will do for a *moderate* time.

The gain in weight from using steel of great strength under high load is not quite so great as some people might think. Thus, suppose a steel axle which will carry safely a load of 12,000 lbs. per square inch is replaced by one which will carry twice this stress. This does not save half the weight of the parts, as the weight cannot be reduced in proportion to the increased stress it will bear, but only in proportion to the square root of the extra stress. Suppose the old axle was 1½ inches in diameter and the working stress per square inch is double, the diameter can be reduced to 1·2 inches. If it is 5 feet long it will weigh 30 lbs. in the first case and 20 lbs. in the second, while the casings of the axle, &c., will weigh as much as they did before. The fact is that the weight of a car is much less dependent on the forgings and axles, such as can be made of special steels, than on the casings for the gear, &c., which are castings, and various accessories, such as radiators, bonnets, &c.

As a matter of simple prudence, tests of reliability should have a preponderating influence over laboratory tests alone. Steels which will stand the greatest stress in tests are by no means necessarily the best for motor cars, since, although of superior strength at first, such strength may rapidly lessen. A metal suitable for a racing car, because it is strong for a few races during a short period of time, may be quite unsuitable for a car from which a service of several years is expected. This is shown by experience. For instance, makers of racing cars use steel which would not be accepted at all by such institutions as Lloyd's, the Admiralty, Board of Trade, &c., to whom safety, in long continuous use, is of prime importance.

As the actual weight of the forgings, in which weight can be saved by the use of special steels and working to higher stresses than allowed in other engineering, is such a very small proportion of the total weight of the car it seems better not to do this, but rather to save weight by making all castings, casings, &c., as compact and light as possible. This applies more especially to axles, &c., the failure of which may cause a serious accident. This will, of course, be subject to modification by practical experience, but, as mentioned, such experience must be of considerable duration on enough pieces to prove that the material can commercially be made uniform.

As steel gear wheels have been so little used in ordinary engineering practice, and as the experience with them has been so short in motor car work, the factor of safety allowed must be ample. The tendency is to make the gears larger in order to avoid the excessive repairs formerly needed. As the drive is usually now direct, gears are so little used that there should be no difficulty in making them to last as long as the rest of the car. Bevel gear is now also made large enough to give a good margin of wear and ample strength.

Castings must generally be rigid, therefore their strength is ample, especially as the thickness cannot conveniently be less than $\frac{3}{16}$ inch. The usual stress allowed for cast iron is 2,500 lbs. per square inch, while 3,000 lbs. per square inch is considered to be a high allowance.

Aluminium is credited with a breaking stress of about 12 tons to the square inch with a fair elongation, but actual experience shows that it will not safely bear a higher stress than cast iron. This is probably

due to its low elastic limit, for when this is passed breakage occurs sooner or later.

Gunmetal or brass is not much used in car construction for parts subject to much stress.

Malleable cast iron is often used, especially for the bevel gear boxes on live back axles, hubs for wheels, &c. It varies a good deal, both in strength and toughness, and is best for thin castings. Generally speaking, it is safe to allow a load about two to three times that allowed on cast iron. Malleable cast iron holds studs well, while gunmetal of all kinds and aluminium are not very trustworthy in this respect.

If very light and strong castings are required, and aluminium is not used, the best material will probably be one of the strong bronzes. There are a large number of these in the market which can be cast with a strength of some 30 tons, and will carry a load in many cases as high as forged mild steel with safety.

There are many parts which have room for shafts, &c., of ample strength, but good wearing qualities are important. In this case a very much harder steel will be employed than is usual in ordinary engineering.

In crank shafts and the shafts of gear boxes this is especially the case. If possible these should be case-hardened, and ground after hardening, as this reduces the wear on the bearings very much indeed. On the other hand, case-hardening probably reduces the resistance of steel shafts to shock owing to the greater liability of the surface to be cracked; once a crack is started it will probably soon extend right through the shaft. As case-hardened shafts or pins must be rigid for good wear, such shafts, &c., are generally amply strong enough whatever steel they are made of.

If ball bearings are used, a hardened sleeve on the shaft generally takes the wear, and this might possibly be done with plain bearings when shafts have a great stress to carry.

The question of margin of wear is analogous to that of safety. Bearings may be made of any suitable size within certain limits; the smaller they are the lighter will the machine be, while the larger they are the longer they will last. A machine can be made with bearings large enough to last for years in daily work without adjustment, provided they are properly lubricated, but this is not practicable with such a machine as a motor car on account of the weight. In fact here, as in the case of the question of dimensions of parts, stationary practice differs very much from that of marine and locomotive practice owing to the difference of weight. In many cases ball bearings are used and here engineering practice does not materially help, except that the usual rule must be followed, which is that if they are to wear well they must be of ample size for the work they have to do. Ball bearings which are too small for their work are an endless source of trouble. If plain bearings are used the ordinary rules in engineering for their size depend on allowing so many pounds pressure per square inch of bearing surface, but such rules are not good if the large bearing surfaces are obtained at the expense of rigidity, or of putting the bearings eccentric to the loads they carry.

In the case of engine bearings we are in a very different position to that of a steam engine, as the working pressure only comes on the bearings every other revolution and the greatest pressure only for a very short time, just at the explosion point. Taking this initial pressure as a standard, it seems usual in gas-engine practice to allow about the following pressures, the

bearing surface being taken as the length multiplied by the diameter:—Gudgeon pins, 3,500 lbs.; crank-pin bearings, 1,500 lbs.; main bearings, 700 lbs. These are, of course, for slow-running open engines with ordinary drip lubrication. Petrol motors run very much faster than these, but, on the other hand, are not often run at their full power continuously, are run with special lubrication, and do not have to run so long without adjustment as stationary engines. It is found that bearings which are not loaded to pressures exceeding the above at full power are satisfactory in practice; but these should not be exceeded except in the case of bushes running on a hardened and ground pin (see Chapters vi. and vii.).

There is no difficulty in making the road wheel bearings large enough to keep the pressures down to about 150 lbs. per square inch, and as they are always carrying the load this should not be exceeded, if avoidable. This is about the pressure allowed on railway axle bearings, though in the case of the crank-axle bearings it has often to be considerably exceeded on account of want of room.

What applies to bearings as regards wear applies generally—*i.e.*, we must have a large margin if the machine is to work satisfactorily. For instance, this applies to the ignition. In the first place, there must be a large margin of power. There should be a certainty of the ignition being effected under the most unfavourable circumstances. It is foolish to rely on an ignition which will only just ignite under the most favourable conditions. This is simply a question of putting plenty of power into the spark, and not trying to do with too little electricity.

This may be illustrated by the boiler of a steam engine. If we take two boats, or two locomotives, with the same sized engine, one with a boiler which will just make all the steam that the engine will take when it has the most skilful driving by a man who is very expert, and not the least tired, and using the very best coal that can be got. In fact, that will just keep steam on a trial trip. The other with a boiler twice as big. On a trial trip, with everything just tuned up right, &c., the first will be the faster, as the boiler will be half the weight of that in the other machine, while the power will be the same. In ordinary running, however, when inferior coal has to be used and men are tired and not quite so skilful, the engine with the big boiler will be the faster and more satisfactory, for the one with the small boiler will always be short of steam. This should be borne in mind in the case of cars, and should always be considered in connection with the results of trials. In steam practice the usual plan is to supply boilers about twice the size needed for generating the required quantity of steam under the most favourable circumstances, and often larger than this.

So with the power of a car. There should be ample power for the gear chosen, even when the engine is working very far from its best. The low gear, in particular, should be low enough for the car to go up, say a gradient of at least 1 in 6 with the engine doing only two-thirds of the best power that an expert can get out of it.

In all wearing parts there must be ample margins, particularly in the low-tension make-and-break. This is a very simple matter of testing, as it is always easy to run an engine night and day on a testing bench at full power for a few weeks and anything likely to wear soon becomes evident. All electric distribution contacts, &c., should be well tested in this way before it is adopted as a standard pattern.

In making calculations it must be remembered that rules simply

epitomise the results of experience, and are only applicable so far as that experience is concerned. All rules must be modified by experience. All we can say in making a rule is that past experience has indicated that, say, axles stressed above a certain stress have frequently failed, and that axles stressed below that never have. The strong presumption is the experience of the future will confirm that of the past if all the conditions and circumstances are the same. But it often happens that the conditions and circumstances are not *all* known. If there is any deficiency of knowledge, two sets of experiments or two periods of experience are not strictly comparable. In all cases in making calculations as to a new machine, such as a car, it is well to compare the size of the parts both with ordinary engineering practice and with other car practice, carefully observing whether any departure from ordinary engineering practice gives really reliable results. Direct experiment is the best guide to the size of parts in all cases, but the experiments to be of any use must be of long enough duration to show the effects of fatigue. They should also be on a sufficient number of parts to discover the effects of possible variations in quality of material.

Calculations of Stress.—The ordinary rules for the calculation of stress in machinery are applicable to cars, and are here only dealt with so far as to allow of a comparison being made between the stresses on the parts of any particular design of car with those of ordinary engineering, and between the stresses in different cars.

The simplest way of doing this is to take the principal parts of a given car, allow such stresses as have been assumed to be usual for this class of work, and then determine the sizes of the various parts. In most cases it will be seen that they correspond fairly well, both with the best modern car practice and with ordinary engineering practice. The principal exceptions will be noticed in due course.

For the sake of illustration, let the car weigh 3,100 lbs. when fully loaded, of which 2,000 lbs. are on the back wheels and 1,100 lbs. on the front, and have 32-inch wheels and a ratio of gear from the engine to road wheels on the top speed of 3 to 1. For a live-axle car this means a ratio of gear on the back bevel of 3 to 1. In the chain-driven car the ratio of gear from the back wheels to the cross shaft is assumed to be 2 to 1 and the gear down on the bevel drive $1\frac{1}{2}$ to 1. The engine has cylinders 4 inches in diameter, the length of stroke 4 inches, and the compression, &c., give an explosion pressure of 300 lbs. per square inch. The stresses allowed are 12,000 lbs. per square inch for all revolving parts; 15,000 lbs. for parts which never have the stress reversed, such as fixed axles, &c., when made of forged steel; and 2,500 lbs. for cast iron.

Maximum Stress on Transmission, what will make the Wheels Slip?—It must not be forgotten that the twisting stress on all revolving parts in the transmission is limited by that which makes the back wheels slip on the ground. In a car with a very low first speed the calculated stress may be much higher than the actual if this point is neglected, and where such a gear is provided it is only for use for running very slow in traffic, &c., and to be certain of climbing steep hills, even when the engine is running badly, &c. In any case it is not possible to get a greater twisting stress than that which makes the back wheels slip, as either slip will happen or the car will continue running. Similarly, in the case of a car which does *not* have such a low gear as this, we may still cause the wheels to slip by putting in the clutch suddenly; in fact, this is more likely to be

done in practice than when the low gear is very low, as the higher the low gear the more difficult it is to start the car on a hill.

Therefore, the stress which makes the wheels to slip will be taken as a basis. The coefficient of friction between the tyre and the ground should be known to get this accurately, but can only be surmised. It will depend a good deal on whether non-skids are used, and this, therefore, has a great effect on the durability of the gear, &c. A coefficient of friction of ·4 will be adopted, as it corresponds well with actual practice.

Back Axles—Twisting Moment.—Taking the case of a live back axle first, the twisting stress, in inch-lbs., on this will be

$$T = W \times \cdot 4 \times R,$$

where R is the radius of the wheel in inches, W the weight on it in lbs., and ·4 the assumed coefficient of friction.

For the above car the stress is

$$1{,}000 \times \cdot 4 \times 16 \text{ inches} = 6{,}400 \text{ inch-lbs.}$$

The diameter of a shaft to carry a given twisting stress is found by the formula

$$d = \sqrt[3]{\frac{T}{F} \times 5{\cdot}1}.$$

Should it be desired to find the twisting stress on an axle of known size, it may be inverted, thus—

$$F = \frac{T \times 5{\cdot}1}{d^3}.$$

In these, d is the diameter of the shaft in inches, T the twisting moment in inch-lbs., and F the stress in lbs. per square inch.

The diameter of a shaft to carry the stress found above is then

$$d = \sqrt[3]{\frac{6{,}400}{12{,}000} \times 5{\cdot}1} = \sqrt[3]{2{\cdot}72} = 1{\cdot}4 \text{ ins. (approx. } 1\tfrac{7}{16} \text{ ins.)}$$

This diameter would be the right one if the revolving axle takes no load, as in fig. 266. It will also be right for the inner end of an ordinary live axle.

Bending Moment.—For an axle which carries the road wheel on the revolving axle, there is, in addition to the twisting stress, a bending stress due to the load. This can be found from

$$M = W \times B,$$

where M is the bending moment in inch-lbs., and B = the distance from the wheel track to the bearing in inches.

Suppose the amount the wheel overhangs is $1\frac{3}{4}$ inches, then the bending stress is

$$1{,}000 \times 1{\cdot}75 = 1{,}750 \text{ inch-lbs.}$$

Equivalent Twisting Moment.—It is convenient, in calculating the stress on a shaft which is under combined bending and twisting stress, to combine these and find the equivalent twisting stress. This can be found from the formula

$$E = M + \sqrt{T^2 + M^2},$$

where E is the equivalent twisting moment. Taking the above case, it is

$$E = 1{,}750 + \sqrt{6{,}400^2 + 1{,}750^2} = 1{,}750 + 6{,}640 = 8{,}390 \text{ inch-lbs.}$$

The diameter is then found by the same formula as before, and is

$$d = \sqrt[3]{\frac{8{,}390}{12{,}000} \times 5{\cdot}1} = \sqrt[3]{3{\cdot}57} = 1{\cdot}53 \text{ ins. (approx. } 1\tfrac{1}{2} \text{ ins.)}$$

This need only be the diameter at the outside bearing, while between this and the differential it can be tapered down to a diameter of 1·4 inches.

Bevel Gear Shafts.—For the bevel gear shaft the twisting stress is reduced in proportion to the ratio of gear down, but the weight, W, will be the weight on *both* the wheels instead of one; then

$$T = \frac{W \times {\cdot}4 \times R}{G},$$

where G is the ratio of gear—in this case, 3; hence

$$T = \frac{2{,}000 \times {\cdot}4 \times 16}{3} = 4{,}266 \text{ inch-lbs.,}$$

and by the previous formula

$$d = \sqrt[3]{\frac{4{,}266}{12{,}000} \times 5{\cdot}1} = \sqrt[3]{1{\cdot}81} = 1{\cdot}22 \text{ ins. (approx. } 1\tfrac{1}{4} \text{ ins.)}$$

If the pinion is overhung, there is a considerable bending stress in addition to the twisting stress; but this cannot be easily calculated, as it is largely due to the thrust of the teeth. This being so, it will be greater as the ratio of gear is greater. In practice, the bevel wheel shafts are generally somewhat larger than is indicated above.

The stress on the propeller shaft and square shaft in the gear box are almost pure twisting stresses, though the square shaft also has a certain amount of bending stress from the thrust of the gear, which depends on the length of the gear box. It has also to be very rigid to keep the gears accurately in gear.

For live axles the above formula seems to agree very well with actual good practice, though the tendency in the best practice would seem to be to make the axles slightly larger, so as not to exceed a stress of, say, 10,000 lbs. per square inch. A large margin should be allowed for axles, as the load on a car is often greater than that it was intended to carry, and the results of an axle breaking are serious.

Cross Shafts.—In the case of a chain-driven car, the twisting strain on the cross shaft is reduced in proportion to the gear down. Thus, if G is the ratio of gear, from the road wheels to the cross shaft it is $\frac{T}{G}$.

In above case, where the ratio is 2, it is

$$\frac{6{,}400}{2} = 3{,}200 \text{ inch-lbs.}$$

If any bending stress is excluded the diameter can be found in the same way as before, and is

$$\sqrt[3]{\frac{3{,}200}{12{,}000} \times 5{\cdot}1} = \sqrt[3]{1{\cdot}36} = 1{\cdot}1 \text{ ins. diameter (approx. } 1\tfrac{1}{8} \text{ ins.)}$$

In practice the bending moment is very small if the bearing is brought well out to the chain wheel and the latter is slightly belled so that the chain line comes over the outside end of the bearing. Often, however, this is not the case and then we get a bending moment as well.

To find this the pull on the chain must be ascertained by means of the formula

$$W \times \cdot 4 \times \frac{R}{r},$$

where R is the radius of the road wheel and r that of the chain wheel.

Suppose the latter is 15 inches diameter = 7·5 inches radius. In above case it is

$$1{,}000 \times \cdot 4 \times \frac{16}{7 \cdot 5} = 853 \text{ lbs.}$$

Suppose the chain wheel overhangs the bearing $1\frac{1}{2}$ inches, the bending stress will then be

$$853 \times 1 \cdot 5 = 1{,}280 \text{ inch-lbs.}$$

The equivalent twisting stress is found as before and is 4,712 inch-lbs., and the diameter of the shaft, by previous formula, is 1·26 inches ($1\frac{1}{4}$ inches).

This shows how much greater a long overhang makes the stresses on the cross shaft; the use of very small shafts in some cars does not depend on material which will stand very high stresses so much as on bringing out the bearing well to the chain wheel. The pull of the chain also varies with the size of the chain wheel or the back wheel, and is very small with the large chain wheels often used.

The longitudinal shaft in the gear box will have the same stress as the shaft in the live back axle, the ratio being the same.

The compression stress on the radius rod of a chain-driven car should be allowed for, and is the same as the pull on the chain.

The radius rod also serves to prevent the axle turning, and, therefore, takes the same twisting moment as the live back axle. This can also be calculated from subsequent formulæ.

Fixed Axles.—Front axles and fixed back axles are beams subject to a simple bending stress. This stress is equal from spring seat to spring seat, and diminishes from there to the wheel track. It can be shown graphically as in fig. 310.

The bending moment in inch-lbs. is found by M = W × B, where W and B are as before.

Taking the back axle, let us assume that the centre of the spring seats is 8 inches from the centre of the wheel track. The bending moment is then 8,000 inch-lbs.

The strength of a solid round bar to resist bending is only half its strength to resist twisting; hence, the formula for diameter is

$$d = \sqrt[3]{\frac{M}{F}} \times 10 \cdot 2.$$

In this case it is

$$\sqrt[3]{\frac{8{,}000}{15{,}000} \times 10 \cdot 2} = \sqrt[3]{5 \cdot 44} = 1 \cdot 76 \text{ ins. diameter } (1\tfrac{3}{4} \text{ ins.}),$$

taking F at 15,000 lbs. per square inch. The same calculation applies to the front axle.

Let us assume that the springs in this case are 12 inches from the wheel track, then the bending moment is

$$550 \times 12 = 6{,}600 \text{ inch-lbs., and}$$

$$d = \sqrt[3]{\frac{6{,}600}{15{,}000} \times 10{\cdot}2} = \sqrt[3]{4{\cdot}42} = 1{\cdot}64 \text{ ins. diameter (approx. } 1\tfrac{5}{8} \text{ ins.)}$$

The stress on the axle diminishes from the spring seat to the wheel track. Consequently, we can, in theory at all events, make the axle smaller from the spring seat to the wheel track. In practice many axles are made a good deal heavier from the spring seat to the steering pivot, but the latter and the wheel spindle are made a good deal smaller than the centre part of the axle. If the latter is right (as it is by theory and apparently practice) the former must be wrong and a good deal of weight wasted.

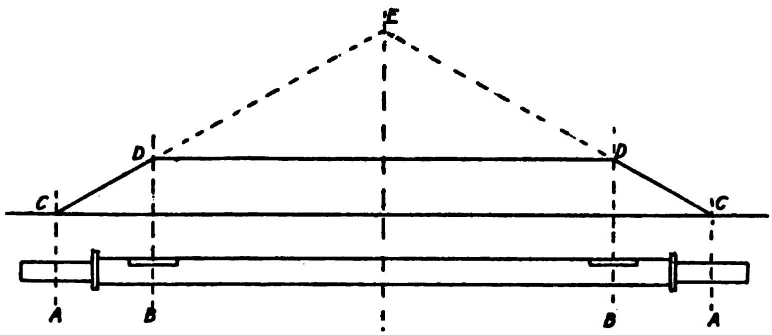


Fig. 310.

A, A, Wheel track. B, B, Centres of spring seats.

The line C, D, D, C represents the bending moment on the axle, the bending moment being proportionate to the height of this line from C, C.

For any different positions of spring the bending moment can be determined by taking the height to the line C, D or a continuation of it.

The height to E would represent the moment, if the whole weight was taken at centre of axle.

The bending stress can be found either by measurement on the diagram (fig. 310) or by arithmetical calculation.

In the latter case, if the wheel spindle in the front axle only extends 4 inches from the wheel track, then the bending moment is

$$550 \times 4 \text{ inches} = 2{,}200 \text{ inch-lbs.,}$$

and the diameter

$$\sqrt[3]{\frac{2{,}200}{15{,}000} \times 10{\cdot}2} = \sqrt[3]{1{\cdot}5} = 1{\cdot}15 \text{ ins. diameter (approx. } 1\tfrac{1}{8} \text{ ins.)}$$

Similarly, if the steering pivot is $5\frac{1}{2}$ inches from the wheel track, and of the pattern, figs. 269 and 270, where it is in bending stress only, the bending moment on it will be

$$550 \times 5{\cdot}5 = 3{,}025,$$

and the diameter

$$\sqrt[3]{\frac{3{,}025}{15{,}000} \times 10{\cdot}2} = \sqrt[3]{2{\cdot}06} = 1{\cdot}27 \text{ ins. diameter (approx. } 1\tfrac{1}{4} \text{ ins.)}$$

If the pivot is of the type of figs. 268 and 271, the pin will be mainly in shearing stress. In this case, if the pin is large enough to give anything like the required bearing surface, it will be amply strong enough. The stress may, however, be worked out; the stress will be

$$\frac{W \times B}{L \times A},$$

where L is the length between the jaws, and A the area of the pin.

In such a case the pin would probably not be less than $\frac{5}{8}$ inch, and the length between the jaws would be about 4 inches at the least. The area of $\frac{5}{8}$ inch is ·3, and the shearing strength per square inch will, therefore, be

$$\frac{550 \times 5{\cdot}5}{4 \times {\cdot}3} = 2{,}520 \text{ lbs. per square inch.}$$

This is ample for safety.

In all cases where it is possible a hollow shaft or axle will be stronger for its weight than a solid one. The size of a hollow shaft of the same strength as a solid one is

$$d^3 = \frac{d_1^{\,4} - d_2^{\,4}}{d_1},$$

where d is the diameter of a solid shaft, and d_1 and d_2 the outside and inside diameter of one of equal strength. In other words,

$$d_1 = \sqrt[3]{\frac{d^3}{1 - x^4}},$$

where x is the ratio of internal to external diameter.

The relative weights of solid and hollow shafts of the same strength, with various ratios of thickness, are as follows:—

Ratio of Internal to External Diameter.	Diameter.		Thickness.	Weight.
	External.	Internal.		
·5	1·02	·51	·26	·78
·6	1·06	·64	·21	·70
·7	1·10	·77	·17	·62
·8	1·20	·96	·12	·51
·9	1·43	1·29	·07	·37
·95	1·77	1·69	·04	·27

Compared with a solid bar having a diameter = 1, and weight = 1.

It will be seen that the weight for a given strength gets less as the diameter is increased and thickness reduced, and this, theoretically, would go on to infinity. In practice, however, there is a limit beyond which the tube cannot develop its full strength, as it fails from local buckling. What actually makes the strongest tube for its weight it is difficult to say, but

in cycle practice tubes are often used whose thickness is only $\frac{1}{30}$ of their diameter. It might be risky without more experience to use such thin tubes as this in motor work, but if the thickness is $\frac{1}{20}$ the diameter (*i.e.*, ·9 ratio) the above front axle would be approx. 2·35 inches diameter and ·065 inch thick (approx. $2\frac{3}{8}$ inches × $\frac{1}{16}$ inch). If thickness = $\frac{1}{10}$ diameter (*i.e.*, ·8 ratio), diameter is practically 2 inches and thickness $\frac{1}{8}$ inch, the relative weights of these compared with the solid can be seen from above table.

For fixed axles I-section steel is often used. This is stronger for its weight than the tube, if the vertical load only is taken, but is not at all strong in any other direction. It has little resistance to torsion, and is relatively weak horizontally. These defects would be a great deal less in square tube than I-section, and this would have the same advantages of vertical strength as the I-section, but it would probably be very inconvenient to make an axle with it. It has, however, been used for frames, and seems an excellent material. There would probably be a difficulty in machining the lugs joining the parts together. With round tube they are bored, but with square tube some more expensive means has to be employed. The calculation for the strength of either channel steel, I-section or stamped steel, or rectangular tube is

$$M = F \times \frac{1}{6H}(BH^3 - bh^3),$$

where M is the bending moment, F the stress per square inch, H the height, and B the breadth, *h* and *b* the height and breadth, less thickness of flanges (fig. 311).

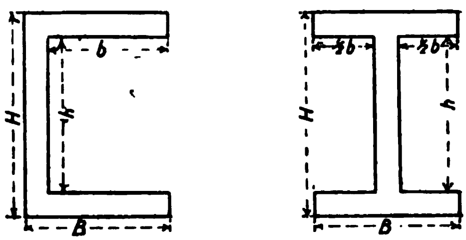


Fig. 311.

The dimensions of this may vary in so many different ways that it is not possible to give any really definite rule like that for round tube; but the strength of different sections will require special calculation.

Take as an example an I-section, having a breadth ·7 of its height, and a thickness of webs ·2 of its height; its strength to resist vertical bending will be

$$\frac{1}{6 \times 1} \times (\cdot 7 \times 1^3 - \cdot 5 \times \cdot 6^3) = \frac{1}{6} \times (\cdot 7 - \cdot 109) = \frac{591}{6} = \cdot 98.$$

That of a solid round bar with a diameter equal to the height of the I-section is ·982, or practically the same; but the weight of the I-section as compared with the solid bar will be

$$\frac{\cdot 4}{\cdot 785} = \cdot 51,$$

or, roughly, one-half.

With these proportions above front axle would be 1·65 inches × 1·15 inches × ·33 inch if of **H**-section, or in round numbers $1\frac{5}{8}$ inches × $1\frac{1}{8}$ inches × $\frac{5}{16}$ inch.

With live back axles the calculation for the tubes of the casing will be the same as that for a tubular fixed axle when there is no tie-rod under the casing. It is common practice to have a tie-rod, in which case the tube becomes a plain strut, and may, theoretically, be made lighter.

The stress on the tie-rod will be $\frac{M}{L}$, where M is the bending stress and L the distance from the centre of the axle to the tie-rod. If this is $6\frac{1}{2}$ inches, in above axle, the load is

$$\frac{8{,}000}{6\cdot5} = 1{,}230 \text{ lbs.}$$

Allowing 12,000 lbs. per square inch working stress, the area will be ·103 and diameter = ·37 inch. In practice this rod is generally screwed, and in this case it would hardly be safe to allow such a high stress as 12,000 lbs.

In practice the sizes of axles and parts as calculated above correspond very closely with actual practice. The tendency is to make front axles rather heavier than above sizes in the case of **H**-section axles. This may be because there are considerable strains on front axles besides those due to the vertical loads. There is a considerable longitudinal strain on the axle due to the resistance of the wheels, especially when they are put over to a great angle in going round corners at high speed, and the **H**-section is very weak in this direction.

Frames.—It is not very easy to calculate what the stresses will be on the side members of a frame, for (1) it varies with the weight and disposition of the passengers, &c.; (2) the body often helps to stiffen the frame; and (3) if there is an inside frame to carry the engine and gear box, it will stiffen the frame as far as it extends. Tonneau bodies probably help a frame a good deal, but side entrance ones can hardly do so at all. An approximation may be made by making a diagram from the separate weights and their bending stresses, and thus drawing a curve of the bending stresses along the frame.

The bending stresses are found from the formula

$$M = W\frac{L_1 L_2}{L},$$

where W is the weight, L the distance between supports, and L_1 and L_2 the distance of the weight from each support respectively.

For a car with a wheel base of 8 feet 6 inches, or 102 inches, as also the weights and distances from the front axle given in the table, the bending stresses will be as follows:—

Nature of Part.	Distance from Front Axle.	Weight.	Bending Moment.
Engine,	24 inches.	$600 = 600 \times \frac{24 \times 78}{102}$	= 11,012
Dashboard, Steering gear, Ignition gear, &c.,	35 ,,	$250 = 250 \times \frac{35 \times 68}{102}$	= 5,835
Gear box, &c.,	50 ,,	$350 = 350 \times \frac{50 \times 52}{102}$	= 8,920
Load,	70 ,,	$750 = 750 \times \frac{70 \times 32}{102}$	= 16,470

Maximum bending moment, 32,750 inch-lbs.

Combining these in one diagram we get fig. 312. In order to get the curve of bending stresses, we add the moments together and produce the outside line, from which we see that the maximum bending moment is about 32,750 inch-lbs. As there are two side plates to the frame, half this comes on each, or 16,325 inch-lbs.

The calculation is then the same as that for the I-section axle. Working this out, we find that a channel steel 3 by $1\frac{3}{4}$ by $\frac{1}{8}$ inches would give us a stress of about 15,000 lbs. per square inch. It will be observed that the bending moments fall to zero at the axles. This would only be so in practice, if no part of the frame overhung the axles, and if the springs were

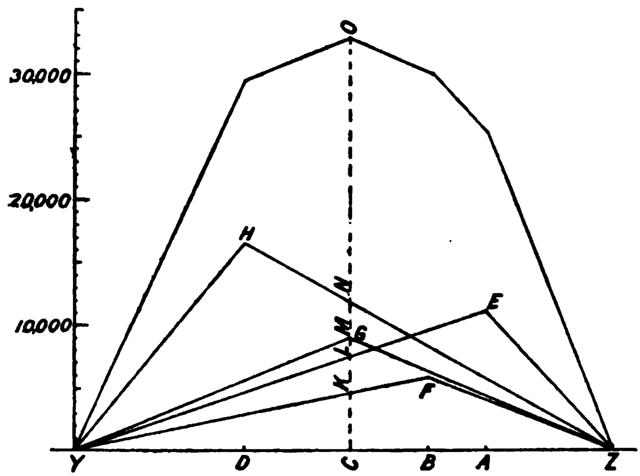


Fig. 312.

Let Z be the front axle, Y the back, Z Y the length of wheel base.

Let A B C D be set off at the distance from Z corresponding with the positions of the weights.

Let heights A E, B F, C G, D H represent the bending moment due to the weights.

The bending moment at any point due to either of these will be proportionate to the height from the line Z Y to a line joining the points E F G H to Y and Z respectively.

The combined bending stress due to all the weights at any point can be found by adding the weights from Z Y to all the lines together, and a curve of bending moment can be drawn as shown.

Thus the bending moment at C is C K + C L + C M + C N = C O and is about 32,750 inch-lbs.

attached to the frame in one place vertically over the axles. The latter condition is only fulfilled when there is a double elliptic spring, and so there are local bending moments between the spring supports. It is, of course, quite possible to get out a bending moment diagram for the whole length of the frame, but it is hardly worth while, as the ends have to be reasonably strong for stiffness. It will be seen from the diagram that when the frame is tapered, as is a stamped steel one, it should be of nearly uniform depth for a considerable way at the middle of the car in many instances.

The depth of the frame will vary as the square root of the bending moment, if the breadth and thickness are constant.

In the case of gears, it would hardly be expected that ordinary

engineering calculations would help us much. Motor-car gear wheels are always made of hard steel, largely on account of the very special method of changing speed, whereas most wheels in ordinary engineering are of cast iron, cast steel, raw hide, or gunmetal. If such wheels could be arranged for, it seems certain that they would run more quietly than the hardened steel ones and be satisfactory if they were large enough, and not required to change gear in the somewhat barbarous manner of the ordinary car. We may, however, take out loads on the gear.

Taking the differential gear first, the load on the teeth of this is $\frac{T}{D}$, where D is the mean diameter of the differential wheel. This is divided among the pinions and must be divided by the number of places in which the pinions engage. If there are two pinions they will engage in four places.

Then for the live axle car above mentioned, the maximum possible load on each tooth, if we assume a 5-inch diameter is $\frac{12,800}{5 \times 4} = 640$ lbs.

The bending stress on the teeth can be calculated from ordinary engineering formulæ, but it seems hardly worth while to do so. One of the largest firms of differential gear makers in France gives a table of differentials which corresponds very nearly to the formula

$$T = d^3 \times 65 + N,$$

where d is the outside diameter of the differential wheel in inches, and N the number of pinions.

By inversion

$$d^3 = \frac{T}{65 \times N},$$

or for the live axle in point if there are two pinions

$$d = \sqrt[3]{\frac{12,800}{65 \times 2}} = \sqrt[3]{98\cdot5} = 4\cdot6 \text{ inches diameter.}$$

In the same way we can take out the stress on the bevel driving gear. The latter is, however, in a different position to the differential, as this only runs when the car goes round a corner, and then but slowly. This is presuming that the wheels are of the same size. As a matter of fact, they are not always so, as some people have a bad habit of using a non-skid on one back wheel but not on the other, which may make a very perceptible difference in their size.

For the main driving gears, however, the size is generally determined by that necessary both for *safety* and for *wear*. If they are large enough to wear well, they are generally large enough for safety. All ordinary engineering rules provide for wear corresponding to some million miles' running of a car, which requires gear of such a size as would be too heavy for a car.

If the differential is of the bevel type, the pins which carry the pinions are usually made of such a size that they have ample margin of safety. They must be of a reasonable diameter, in order to give enough bearing surface not to wear badly. They will be in almost perfect shear, so we need consider this stress only.

The stress will be

$$\frac{T}{R \times A \times N},$$

where R is the radius of the place they are fixed to the cage, T twisting moment as before, N number of pins, and A area of each.

Suppose the cage is 5 inches diameter, or $2\frac{1}{2}$ inches radius, number of pinions, and consequently pins, 2, diameter of pins $\frac{3}{4}$ inch, then for the live axle under consideration

$$F = \frac{12{,}800}{2{\cdot}5 \times 2 \times {\cdot}44} = 5{,}820 \text{ lbs. per square inch.}$$

If the differential is of the face type there is a considerable bending stress due to the thrust of the teeth, which must be allowed for, although it cannot be calculated.

Engine Parts—Bolts and Studs.—In the engine most parts are amply strong, since they are necessarily made large for obtaining good wearing surfaces and rigidity.

The stresses on the bolts and studs should always be moderate as the saving of weight gained by making them small is trifling, while the damage resulting from their breaking may be serious, and they are liable to be unduly strained when screwed up too tight.

In ordinary engineering practice the usual stress allowed for small work of this kind is not more than 5,000 to 6,000 lbs. per square inch.

For main bearing bolts the stress will be

$$F = \frac{P \times A}{B \times N},$$

where P is the pressure per square inch, A the area of the cylinder in inches, B that of the bolts at the bottom of the thread, and N the number of bolts.

If there is a bearing between each crank, with two bolts in each, N may be valued at four. If there is a pair of cranks between each bearing, the load will come mainly on the nearest crank, and this must be allowed for. Allowing 6,000 lbs. per square inch for the maximum stress, then for a 4-inch cylinder with 300 lbs. pressure, we get

$$B = \frac{300 \times 12{\cdot}56}{4 \times 6{,}000} = {\cdot}157 \text{ square inch.}$$

This corresponds to ·42 inch diameter at the bottom of the thread, or a little over $\frac{1}{2}$-inch bolts. In practice, probably $\frac{1}{2}$ inch would be used, which gives a stress of nearly 8,000 lbs. per square inch, but probably the pressure is not often as much as 300 lbs.

If the cylinders are separate, the same calculations apply to the bolts which hold the cylinders on to the crank chamber, but if they are cast together the strain may be distributed over several of the bolts. It is not well to reckon on this, however. For the webs which carry the bearings the same calculation applies, except that the stress should be kept down to about 2,500 lbs. per square inch.

The safe pressure in a cylinder as regards bursting can be found from

$$F = \frac{D \times P}{2\,S},$$

where D is the diameter, P the pressure, and S the thickness

$$S = \frac{D \times P}{F \times 2}.$$

In calculating the bursting pressure of a cylinder from this formula, it must be remembered that at the moment of explosion, when the pressure is greatest, the length of cylinder exposed to it is so short that it is very much supported by the end. Therefore, the cylinder could really be made thinner than calculation shows without the stress exceeding the normal.

In the case of a cylinder 4 inches diameter and 300 lbs. pressure, the thickness is for

$$F = 2{,}500,$$

$$S = \frac{4 \times 300}{2{,}500 \times 2} = \cdot 24 \text{ inch.}$$

The longitudinal stress can similarly be calculated from

$$S = \frac{P \times A}{C \times F},$$

where A is the area, and C the circumference. In above case thickness is

$$S = \frac{300 \times 12 \cdot 56}{2{,}500 \times 12 \cdot 56} = \cdot 12 \text{ inch.}$$

As a matter of fact, cylinders are always made thicker than the calculation shows to be necessary, for practical reasons; a 4-inch cylinder is generally from $\frac{1}{4}$ inch to $\frac{5}{16}$ inch thick.

The stress on the big end bolts is mainly from the inertia of the piston and connecting-rods. Ignoring the effect of the angle of the connecting-rod, and assuming it to be of infinite length, the stress can be calculated from

$$O = \frac{W \times V^2}{32 \cdot 16 \times R},$$

where O is the load in lbs., W the mass in lbs., R the radius in feet, and V the velocity in feet per second.

For a piston and connecting-rod 4 lbs. in weight, having a 4-inch stroke, that is 2 inches radius, running up to 2,000 revolutions,

$$O = \frac{4 \times 34 \cdot 7^2}{32 \cdot 16 \times \cdot 167} = \frac{4 \times 1{,}204}{32 \cdot 16 \times \cdot 167} = 897 \text{ lbs., practically } 900 \text{ lbs.}$$

If there are two bolts in the big end, half this, or 450 lbs., comes on each bolt, and the area required at 6,000 lbs. per square inch is ·075; diameter at bottom of thread, ·31 inch. This would mean $\frac{7}{16}$ inch bolts; that is slightly larger than is customary, but probably none too big for engines which habitually run at such high speed.

Crank shafts have two stresses, the bending and twisting. These can be ascertained separately, and combined as in those on the back axle. In the crank shaft the maximum bending stress will be just at the explosion point, on the dead point when there is no twisting moment. In order to ascertain what is the greatest combined twisting and bending moment on the crank

shaft, a curve of the twisting moment should be made from the indicator diagram. The method of doing this is as follows :—*

"The twisting moment at any position of the crank is equal to the pressure on the piston multiplied by the distance intercepted by a line through the connecting-rod on a line at right angles to centre line through centre of cylinder.

"Let A B (fig. 312A) be the centre line of the engine through the cylinder and shaft centres, A C the position of the crank, B C the connecting-rod, and A D a line at right angles to A B. Produce B C to cut the line A D, and drop from A a line A G perpendicular to B C. P is the load on the piston, and R is the thrust on the connecting-rod. It will easily be proved that the angle D A G is equal to the angle A B D, called for convenience α. Then

$$P = R \cos \alpha \text{; and } AG = AD \cos \alpha.$$

$$\text{The twisting moment} = R \times AG = R \times AD \cos \alpha = P \times AD."$$

Where there are accurate indicator diagrams available they will, of course, be used, but in their absence we must assume pressures for different parts of the stroke.

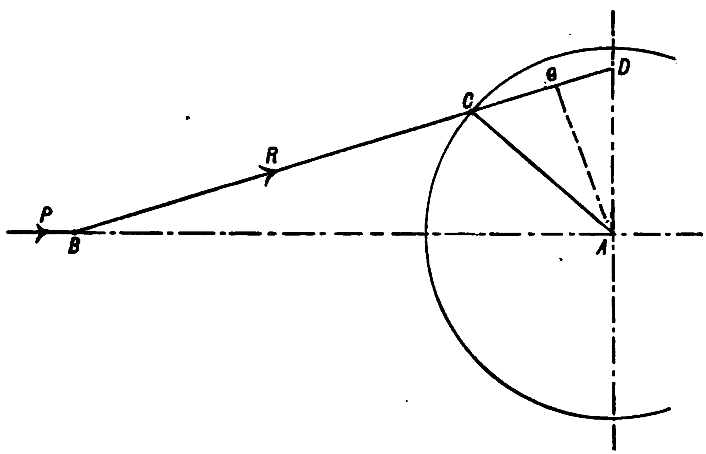


Fig. 312A.

The bending stress on a shaft with two supports and the weight in the middle, is found as follows :—

$$M = \frac{W \times D}{4},$$

where D is the distance the supports are apart. This applies to a crank shaft with a bearing between each crank, but where there is no bearing between the adjacent cranks the bending stress can be found in the same manner as that on frames. The length of A D for the twisting stress can be got most simply by measurement from a full-sized drawing, and from this and the bending stress we can calculate the equivalent twisting stress.

Assuming a 4-inch cylinder, 4-inch stroke, crank shaft with a bearing

* A. E. Seaton, *A Manual of Marine Engineering*, 15th ed., p. 182.

between each crank, bearings 3 inches apart, and connecting-rod 10 inches long, the stresses for each eighth of the stroke will be as follows :—

Point of Stroke in Eighths.	Assumed Pressure. Lbs. per Square Inch.	Total Pressure on Piston.	Length of A D. Inches.	Twisting Moment. Inch-lbs.	Bending Moment. Inch-lbs.	Equivalent Twisting Moment. Inch-lbs.
0	300	3,768	0	0	2,826	5,652
1	230	2,889	1·44	4,160	2,167	6,867
2	160	2,010	1·85	3,719	1,508	5,398
3	120	1,507	2·05	3,090	1,130	4,572
4	100	1,256	2·06	2,587	942	3,692
5	80	1,005	1·92	1,925	754	2,724
6	65	816	1·75	1,428	612	2,165
7	50	648	1·26	837	486	1,456
8	30	377	0	0	283	566

From this a curve can be made if desired.

The pressures are, of course, assumed, but are probably higher than are generally obtained in practical work.

From this it will be seen that the maximum equivalent twisting stress is 6,867 inch-lbs. From this we get the diameter of the shaft by previous formula :—

$$\sqrt[3]{\frac{6,867}{12,000} \times 5{\cdot}1} = \sqrt[3]{2{\cdot}92} = 1{\cdot}43 \text{ inches diameter (approx. } 1\tfrac{7}{16} \text{ inches).}$$

This is ·36 of the diameter of the cylinder, and, therefore, fairly agrees with the usual practice as given in Chapter vi.

The above formula takes no account of the inertia of the moving parts and this is in practice hardly necessary. The effect of it is to reduce the effective pressures at the beginning of the stroke and to increase those towards the end, and this will reduce the actual maximum stresses on the shaft. Considering this and the fact that engines are generally run a great part of their time throttled, it is evident that the ordinary working stress of crank shafts of the usual size is well below the 12,000 lbs. per square inch.

The pressures on the bearings are easily calculated. The area of a bearing is always taken as the diameter multiplied by the length, and the pressure divided by this gives the pressure per square inch.

Thus, if with a 4-inch cylinder the gudgeon pin is ½ by 1½ inches, crank pin 1½ by 2 inches, and main bearings each 1½ by 2 inches, the areas will be ·75, 3, and 6 square inches respectively.

If we assume a pressure of 300 lbs. per square inch, the pressure per square inch on the bearings will be 5,000, 1,252, and 626 lbs. per square inch respectively. The pressure on the gudgeon pin is very much more than is in general use, even for gas-engine practice, but the others are not unusual.

Actual Working Stresses.—By comparing the above calculations with actual practice in cars, it can be seen if the stresses in cars are more or less than those allowed in other engineering.

Generally speaking, there is a pretty close agreement. In most cases,

however, the tendency would seem to be to make such parts as are subject to reversal of stress slightly larger than the above calculation gives, and perhaps it would be safer for the stress to be 10,000 lbs. instead of 12,000 lbs. per square inch. This would apply particularly to the cross shaft of the chain-driven car and the revolving axle of the live-axle car. In most other cases the size of parts follows the calculation very closely, if allowance is made for the side thrust of gears.

One point must be considered in connection with the bending moments on axles near the wheel track. It will be seen from fig. 310 that the moment falls to zero at the wheel track. This is not so in practice, as there are side strains on the tyre of the wheel when going round a corner, and also when one side of the car is higher than the other. Thus the above formula cannot be accepted absolutely if there are very small overhangs. With ordinary amounts of overhang, a calculated stress of 10,000 lbs. will allow sufficient margin for these strains, but all formulæ must be applied with a little common sense.

In actual practice, for instance, a mild-steel back axle, $1\frac{5}{8}$ inches, is found amply strong enough for carrying 2,000 lbs.

For the reasons just given a slight allowance must be made if the front wheel spindle has a very short overhang. The size given by above calculation has, however, been used in practice with excellent results.

Fixed axles are often somewhat smaller than the calculated size—*i.e.*, they are worked to higher stresses than 15,000 lbs. On the other hand, many makers are now putting in I-section axles of such a size that the stress is well below this amount.

The stresses of frames are also kept very low, at all events in the pressed steel type. The car for which the calculation is made has a moderate wheel base; with longer wheel bases the bending moments will be greater, but, even allowing for this, frames are generally made heavier than calculation shows to be necessary. This raises a doubt as to whether the pressed steel frame is really as light as the channel steel of equal strength. Careful calculations based on actual data would indicate smaller stresses than those assumed to be necessary in the foregoing calculation, as the assumption that the weight of each part is concentrated at its centre of gravity is not strictly true.

As regards the transmission gear, calculation and practice agree very well, except that the cross shafts for chain drives apparently seldom have as high a stress as 12,000 lbs. allowed on them. In the case of longitudinal shafts carrying overhang bevels something has to be allowed on the above sizes for the side thrust of the teeth.

In all the above cases of actual practice, it is assumed that ordinary mild steel of about 40 tons tensile strength was used; an assumption fully justified by the fact that for the sizes given steel of this kind has provided an ample margin of safety. If steel is used which is found to be capable, *in practice*, of standing heavier stresses than such mild steel, of course the calculation will be modified accordingly; but, as mentioned, so far the evidence that any steels will do so is small.

CHAPTER XX.

SPECIAL CHANGE-SPEED GEARS.

Special Change-Speed Gears.—It has been somewhat of a surprise to many engineers that the sliding-tooth system of change-speed gears has stayed in use so long, and has, in fact, so far, superseded every other form that has been tried against it. At first sight it seems a brutal thing to change speed by sliding other gears into mesh, and forcibly jerking the clutch on to the speed required. There is no doubt that in many cases it causes a good deal of wear in the edges of the teeth, and that it has many disadvantages. These are not so prominent as they used to be, on account of the fact that gears are wider, and the hardening and tempering of them better understood, and also that with the greater engine powers now used they are not changed so often; and, moreover, with the "direct drive" now so commonly used the drive on a great part of the running does not go through the gear at all. The advantages of the system seem to be very largely in the method of driving rather than in the mechanical construction. The plan of having the speed changed by a positive drive with a lever, and having one friction clutch to drive the whole of the speeds is very convenient. The brake and clutch pedals are close to the feet, so that when stopping or slowing-down, &c., the hand need not be removed from the steering wheel. In fact, as long as the driving is in the same gear it can be entirely managed by the feet. Further, in stopping suddenly it is hardly possible to stop the engine, as instinctively the foot is put on the clutch as well as the brake, and so disconnects it. On the other hand, when driving with belts or clutches worked by hand, and stopping in a hurry, the clutch may not be taken out; consequently the engine will be stopped, and the driver has to get out and restart it. No doubt this is rather a matter of habit, and there are many small cars running well in which the clutches are worked by hand, with great satisfaction to their owners. For large cars, however, the objection to having to constantly take the hand off the wheel is a much greater objection. Other advantages of the standard change-speed gear are that it is of simple construction and easy to understand. It is easy to renew the parts that wear most, and these are not very expensive. It is light.

The disadvantages are the wearing of the teeth in changing gear. The gear cannot practically be made very large for its work, as this would need a gear box of intolerable length. The gears must be made of hard steel, which is not the best material for running quietly.

Whether the sliding-tooth gear will or will not stay in use for commercial work remains to be seen. It certainly does not seem to be durable compared with ordinary gears in other kinds of engineering, which often run every day for many years without being renewed. On the other hand, it is quite possible that it may pay to renew them often, in order to keep the weight down, and that, in this case, the saving in tyres or other expenses will be less than the cost of renewing gears.

There are several unusual forms of gear which are, or have been, in considerable use. Two are the Govan gear fitted to the Argyll cars, and the Renault. The former is shown diagrammatically in fig. 313; as will be seen the second speed is always in mesh, the first being the only one slid in, while the reverse is put in facewards by a separate lever. The three positions for the three speeds are effected by a lever working in a T-shaped slot, so arranged that the second and third are held out of gear positively when the first is put in.

Figs. 313 to 316 are drawn to the same scale as figs. 226 to 234 for the same width of gear wheels and distance of shaft centres; the space occupied by all these different arrangements can, therefore, be compared.

The Renault is shown in fig. 314, from which it will be seen that the countershaft is on an eccentric, and so arranged that the gears do not

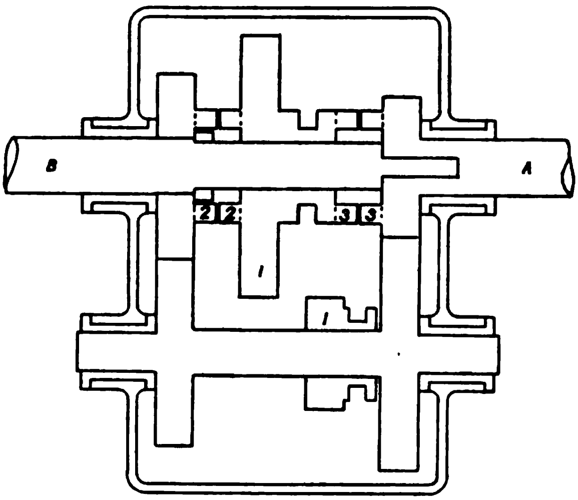


Fig. 313.

To put first speed into gear the small pinion marked 1 is slid along the back shaft, which is square, till it gears into the large wheel 1.

To put the second or third into gear the sliding piece on the main shaft, consisting of the first speed gear wheel and second and third speed clutches, is slid either backwards for the second speed or forward for the third till the clutches engage.

The reverse is by a wide pinion moved into gear with the two low speed wheels by a separate lever.

mesh while the sliding part is being slid along. This requires a very short gear box, as there need be no clearance between the different gears, while in an ordinary run-through gear box there must be enough clearance between each pair of gears for the one to slide out before the other slides in. The countershaft is also entirely out of gear when the direct drive is in use.

Both the above have the usual friction clutch between the engine and gear box, and drive in the usual way.

Several other plans have been used. Several gears have been brought out in which all the wheels were always in mesh, and the gear was changed by sliding a key along a shaft somewhat as in fig. 315. This is difficult to

make satisfactorily, as it is not easy to give the wheels a good bearing on the shaft, and at the same time a good grip of the key. But as very wide gears can be put into a very small gear box it may be successful in the future.

In order to avoid these difficulties I once made the gear represented in fig. 316. This worked very well and the gears were made quite successfully of bronze running on steel. The sleeves have a good long bearing and, therefore, keep the wheels rigid sideways, and it can easily be made for direct drive if desired. The clutches on the spindles are, however, rather expensive to make, and it did not seem to have enough advantage over the ordinary gear to be worth further consideration.

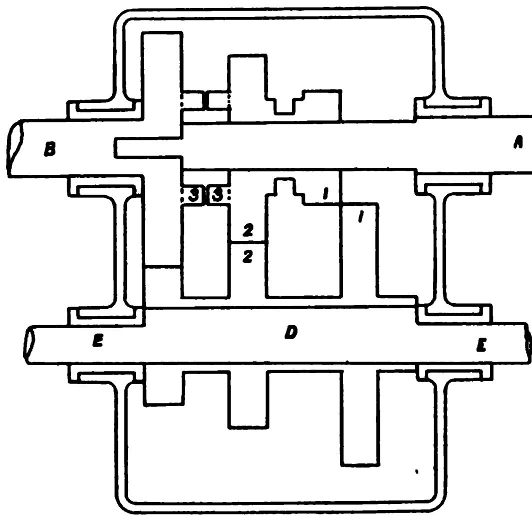


Fig. 314.—Special Change-speed Gear.

As shown, the gear box has the second speed in gear.

To change to the third the shaft, D, is revolved on the bearings, E E, till the gears are out of mesh and the sliding part moved back till the third speed clutch claws engage.

To change to the second or first the shaft is revolved as before and the sliding part moved till the second or first speed wheels are opposite, when the shaft is again revolved till the gears mesh.

For reverse, the sliding part is moved right forward till the first speed wheels clear each other, and a broad pinion moved across them.

All the movements are performed by a single movement of the change-speed lever, which moves straight on as in a run-through gear.

In all gears with the wheels always in mesh, it is necessary to take care that the clutches or sliding parts are on the *driven* and not on the *driving* gears. The reason is explained when discussing this matter with regard to gear boxes in Chapter xiii. In a gear like the last mentioned, if the clutches are on the driven part the friction of the gear is very small, as all the wheels revolve slowly.

If the sleeves go through the bearings also, each sleeve is going slower than the last, and all in the same direction. On the other hand, if the clutches are on the *driving* part each is going faster than the other, and the one that goes the fastest is the outside one which runs against the bearing.

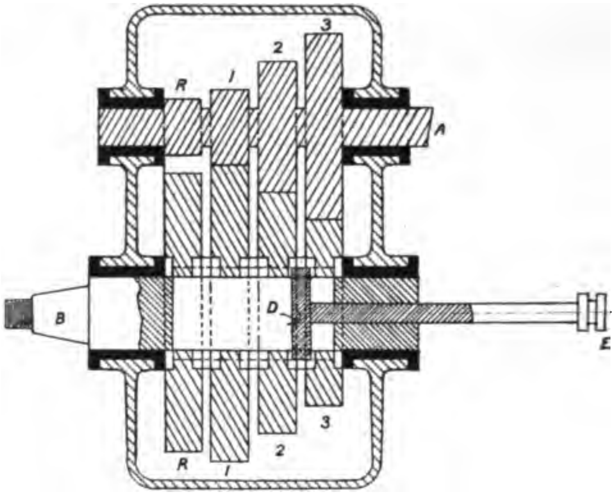


Fig. 315.

The first, second, and third driving wheels and reverse are all secured to the shaft, A, and gear into corresponding wheels running loose on the shaft, B. These wheels have key ways cut in them to fit the key, D, which slides in a slot in the shaft, B, and is moved by the collars, E. These, if moved forward from the position shown, will make B revolve with the third speed; but, if backward, with the second. The difficulty generally is to give the wheels enough bearing on B to be rigid sideways. As shown, collars are pinned on the shaft between the wheels to help this. The key also occasionally breaks. It is not easy to arrange this for a direct drive.

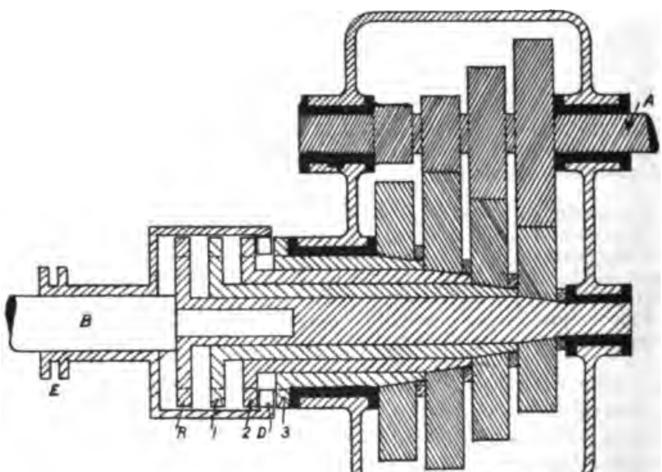


Fig. 316.

In this gear wheels secured to A drive wheels on sleeves, which have on them the claw clutches, R 1, 2, 3 of the reverse, first, second, and third speeds respectively. The clutch, D, is slid along the squared part of B by the collars, E, to engage these as desired.

All the above are variations of the ordinary drive, in which the change-speed is positive and there is one friction clutch for all the speed. There are several others in which there is a clutch of some kind to each speed.

Crypto Gears.—One of the most used of these is the Crypto gear. If in a differential gear of the ordinary bevel type, one bevel wheel is fixed and the other rotated, it will be evident that the cage will revolve at half the speed of the bevel which revolves. We thus get a reduction of 2 to 1. This is not a very convenient ratio for reduction, as a rule, as the Crypto is generally used in cars having only two speeds, in which case the ratio must be more than 2 to 1. In order to get a greater ratio the

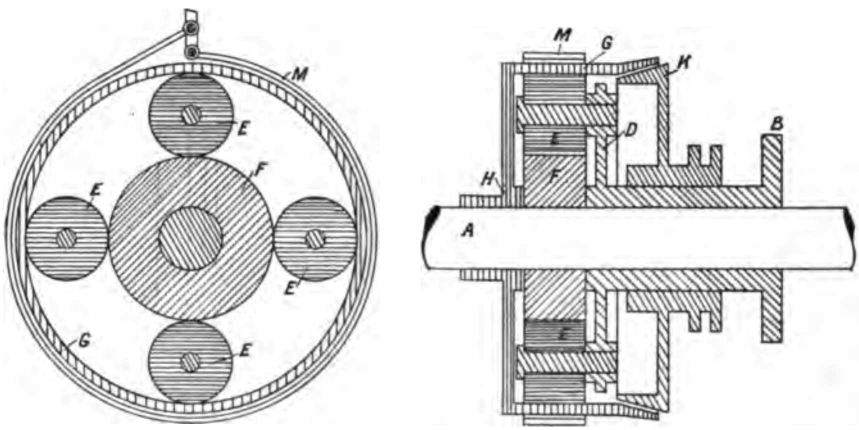


Fig. 317.

This is shown as driving a chain wheel for direct chain drive to a live back axle, as is usual in the class of cars in which Crypto gears are most used. A is the engine crank shaft, and B the chain wheel which revolves with the frame, D; this carries the pins on which the pinions, E, revolve. These pinions gear with an external wheel, F, keyed on the crank shaft, and an internal wheel, G, carried on a frame, H, which runs loose on the crank shaft, A. In the position shown there is no speed in gear, as the wheel, F, will make the pinions, E, revolve, and G will revolve in the reverse direction to F. If, however, the cone clutch, K, is made to engage with G, then the whole gear revolves together, and B is driven at the same speed as the engine. If, on the other hand, K is left free, and the brake band, M, tightened on G, the latter is stopped, and the pinions, E, roll on it and carry the cage, D, at lower speed.

The same gear can be used as a reverse if B is attached to G and freed from D, and the latter held from revolving; but this entails having numerous clutches. Otherwise a separate train of pinions must be used.

driving wheel must be smaller than the fixed wheel. This can be managed by using a bevel with the spindles not at right angles, or an internal and external wheel, or a gear of the type of the face differential (fig. 256), but with the driving wheel smaller than the fixed wheel. The internal and external gear has the advantage in many ways, but, unless the ratio of reduction is very great, the pinions run very fast when on the low speed.

In this type of gear it is usual to have two friction clutches, one to clutch the driving part to the cage, in which case the whole gear goes round as a solid block, and the other to lock the fixed part, in which case low speed comes into use (fig. 317).

Comparing the Crypto with the ordinary gear, the latter seems very

much the cheaper to construct, if more than two speeds are required, and probably cheaper even for two speeds and reverse.

The Crypto requires at least four gear wheels for two speeds only, and these run on three or more centres. There are also two clutches. Four gear wheels on two centres, one clutch and one change-speed lever are all that is necessary in the ordinary type of change-speed gear for two speeds; and each extra speed only requires one pair of gear wheels, whereas a Crypto generally requires a complete chain of wheels and a clutch. It is true the gears in the Crypto are always in mesh, but so would other gears be if they had separate friction clutches.

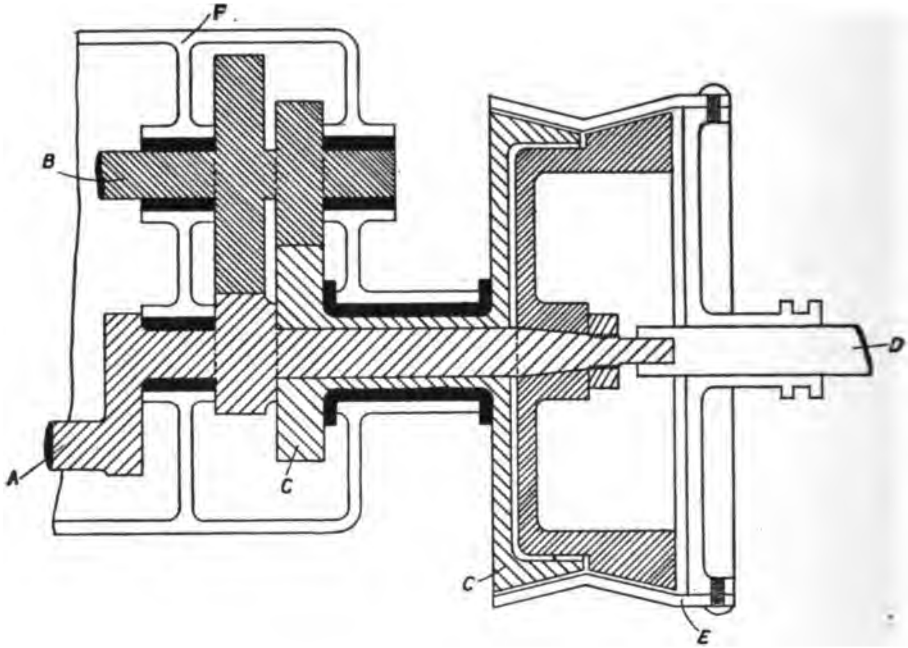


Fig. 318.

A, Crank shaft.
B, Cam shaft.
C, Gear wheel sleeve and cone of low speed.
D, Shaft to chain or other driver.
E, Clutch sliding on D, which engages in either high or low speed.
F, Crank casing.

That the Crypto runs very well on many cars, when well made, is undoubted, but it shows no superiority over the ordinary ones.

Other gears with two clutches can be so arranged that the movement of a lever one way puts one gear into action, while the opposite movement actuates the other, and so becomes an ordinary back gear with two friction clutches. A difficulty here is that, generally speaking, the diameter of the clutches must be smaller than twice the distance between the shafts, which makes them rather delicate. Fig. 318 shows how this may be avoided by the gears being put inside the crank case and the clutches outside. The back shaft then drives the cam shaft, and there are only two extra gear

wheels (besides those necessary to run the cam shaft) in the whole change-speed gear. The gears are carried on the same centres as the cam shaft and crank shaft, and, therefore, there is no extra setting to the crank case and no separate gear box. If oil is kept off the low-speed clutch, this gear works well and should be very cheap to make.

Belt drives are almost obsolete except for motor bicycles, but had many advantages for very light cars. They were very cheap, while no friction clutch will take up the power more sweetly than a belt with fast and loose pulleys. The best pattern was probably that of the old New Orleans, in which there was a single belt running on a wide pulley at the engine end; and on the back axle, a countershaft having two fast pulleys connected respectively with two different pinions gearing on to the back axle with two different ratios of gear. Between these pulleys there was a loose pulley on which the belt ran when the car was at rest, while it could be shifted on to either of the others for the low and high speeds respectively. As there was only a single belt there was no difficulty in providing an adjustment for keeping it tight, and as it ran the whole length of the car and could be made reasonably wide it did not give much trouble from slipping.

The whole question of special change-speed gears may, in the future, require a good deal of consideration, as, although the present form does very well for pleasure cars, it is by no means perfect. It entails a good deal of wear in the edges of the teeth even with very careful driving and the best case-hardened wheels. Then it entails hardened steel gear wheels which are by no means the best for running quietly. It is also liable to great damage by bad driving. These defects are not nearly so prominent in pleasure cars as in commercial vehicles, principally because the former are seldom driven enough sufficiently long and often for them to be very apparent. Another reason is that in the modern pleasure car the top speed has a direct drive on which most of the running is done. In most commercial vehicles, on the other hand, there is a good deal more changing of speed owing to the greater traffic in their routes, and as the direct drive is not so much used the gear becomes very noisy.

Another case in which special change-speed gear may be useful is in the very cheap, light car. So far there is nothing in general use between a car with all the parts of the large car in miniature and a tricar with only three wheels. The latter is much the cheaper and lighter mainly because it does not have the same arrangement of change-speed gear as in the car. The advantages of four wheels over three are very great, and there should be room for a really light and cheap car to compete with the tricar. This might probably be made most satisfactorily with some special change-speed gear.

Should a large engine and two speeds become general for cars, a different kind of gear from the ordinary arrangement of the present time will probably be introduced.

CHAPTER XXI.

SPECIAL TYPES OF CARS—COMMERCIAL VEHICLES.

Special Types of Cars.—Although the type of car with the engine vertical in front holds the field to a very large extent at present, it is by no means perfect; it is quite possible that in the future it may be superseded by some other type; but, if so, it would be rash to predict what the type will be. There are, however, several interesting types being now made which have points worth considering.

Defects of Ordinary Type.—The great fault of the ordinary type of car is the length taken up by the bonnet, which necessitates the body being made to project beyond the back wheels. The first very much lengthens the car and consequently increases the weight of the framing, &c. It also makes it much less easy to turn in a small space. It is quite true that, on the other hand, lengthening the wheel base has made the cars more comfortable to ride in. It is doubtful, however, whether such excessively long wheel bases as are sometimes used now are necessary for comfort, if the body came well between the wheels. In all other vehicles it has been found that the smoothest running is obtained when the weight is concentrated between the wheels. Any one who will look at an ordinary carriage of any kind will see that the back seats are, in most instances, well in front of the back axle. On the other hand, with a four-seated car having the engine in front the back seats always overhang the axle, generally a good deal. The effect on the comfort of the passenger will be seen by considering what happens when the back wheels go over a bump. If the seat is midway between the wheels it is only lifted *half* the extent of the bump, if exactly over the back axle it is lifted the *whole* extent, and if it is behind the back axle it is lifted *more* than the height of the bump. The best place for the seating accommodation, therefore, should be as nearly as possible halfway between the wheels.

If there is a long bonnet in the front of the car it is impracticable to place the back wheels so far back that the seating accommodation is anywhere near midway between the wheels as this would make the wheel base excessively long, and also the greater part of the load would be on the front wheels, and the back would have an insufficient amount on them to make them grip. In practice, therefore, the back wheels are always placed nearly under the seating accommodation, so that the weight of the load is carried on the back wheels and that of the engine on the front. This necessarily means that the passengers feel the inequalities of the road a great deal more for the reasons given. It has an additional defect—viz., that it makes the car a great deal more liable to side-slip (see figs. 319 and 320).

In order to get the steadiest running vehicle, and the one least liable to side-slip, the weights should be well in the middle between the wheels. In fact, the more nearly the weights are concentrated at the centre of gravity of the whole vehicle the less will be the effect of their inertia.

The effect of putting weight at the ends of a vehicle is, in fact, exactly the same as that of taking weight from the centre of a flywheel and putting it in the rim, and, therefore, the greater the weights at the ends of a car the more force it will take to set them in motion and the more to stop them when started. On the other hand, the power of the wheels to control this increases with their distance apart, so that we can, to a certain extent, counteract the effect of the bad distribution of weights by increasing the wheel base, but if we could concentrate the weights we should get better running than the present long cars with shorter wheel base, and handier and lighter cars. The effect of inertia is seen both vertically in the tendency vehicles heavily weighted at the ends have to lurch and pitch,

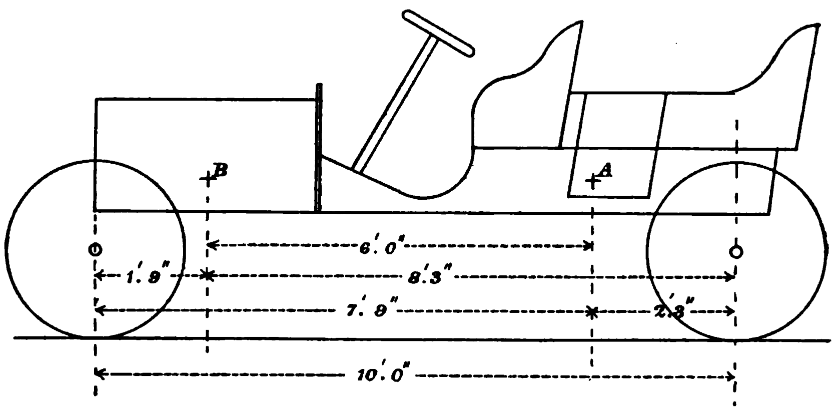


Fig. 319.

Fig. 319 shows a car with a bonnet 3 feet 6 inches long, and a side entrance body similar to fig. 307. A is the approximate centre of gravity of the body loaded, B that of the engine. The position of the front axle is fixed by the length of the bonnet, so that it is 7 feet 9 inches forward of the centre of gravity of the car body. In order to have the latter midway between the wheels, the back axle should be 7 feet 9 inches behind this, making a wheel base of 15 feet 6 inches, which is, of course, impracticable. In practice a 10 feet 6 inches wheel base, as shown, is inconveniently long, and this brings the back axle right under the back seats, which partially overhang. The centre of gravity of the body and engine are 6 feet apart, which make their rotary moment of inertia considerable. Further, 75 per cent. of the weight of body and passengers is carried on the back wheels, and 80 per cent. of the weight of engine on the front. In consequence of this, the proportion of weight which will be on the back and front wheels will vary very much with the weight in the car, and if the proportion is right for one load it will not be right with any other.

and also in side-slip, the main cause of which is the tendency of the car to continue rotating when once started, and the inability of the wheels to stop it.

The above considerations will show why it is bad to shorten the wheel base by letting any part of the engine and bonnet overhang the front axle. It also explains why the vertical engine arrangement suits the racing car so well. In this the whole car is little more than an enormous engine on four wheels, and the engine is placed so far from the front axle that it is really well in the middle of the car.

Theoretically, the proper plan would be to shift the engine to the middle of the car, and place the body over it, but this would render the engine in-

accessible. The objection would be removed if the engine did not need frequent attention, a condition which may ultimately be realised.

The most obvious alteration in the general arrangement is to put the driving seat over the engine in place of the bonnet. This has sometimes been done in pleasure and commercial cars, but not generally; but it remains to be seen if it will become the usual practice when the difficulty of making the engine accessible and of conveniently arranging the connections to the change-speed gear has been overcome. A slight modification of the ordinary design, sometimes adopted, is to make the floor of the front seat very sloping, and to put the engine partly under it. In this case the bonnet is somewhat shortened, but the engine is rather less accessible.

The next variation is to put the engine under the floor of the front seat, and have an engine of the type of fig. 118. This is adopted by Messrs. Rolls, and makes a much more satisfactory design, as the valves, ignition

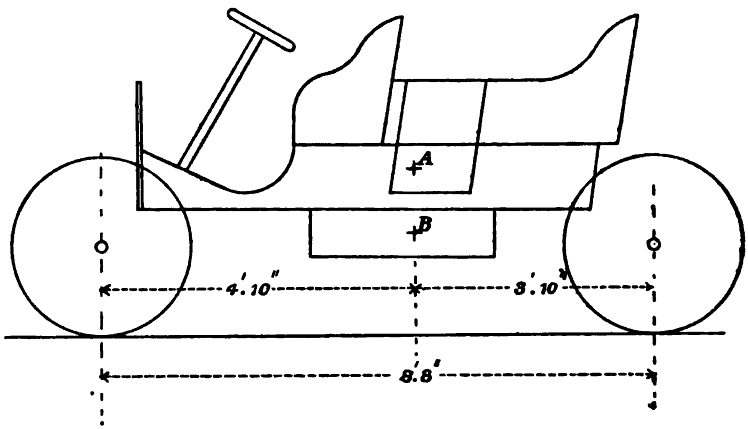


Fig. 320.

Fig. 320 shows the same body, but arranged with the engine underneath it, the centre of gravity of the engine being exactly under that of the body loaded. The wheel base is shortened by nearly 2 feet, yet the wheels have far more control over the weight as regards side-slip. The centre of gravity of the body and engine are both nearly midway between the wheels, and the proportion of load on the front and back wheels is not disturbed by the amount of the load.

plugs, and all parts of the engine which need be accessible are well at the side.

The majority of designs which depart very much from the ordinary have horizontal engines. Much has been said for and against these, and the advocates of the vertical engine have contended that the weight of the piston on the side of the cylinder will wear it oval. This is, of course, absurd, as the weight of the piston in any petrol motor is so slight that it will produce no perceptible wear. On the other hand, the thrust of the connecting-rod is considerable, and if any wear takes place it will be from this. Further, in a horizontal engine it is possible to arrange that the thrust comes on the bottom of the cylinder, where the piston will be thoroughly well lubricated.

That the horizontal engine has, on the whole, the greatest advantage for all ordinary work seems, on the face of it, to be evident from the fact that it

is generally employed if there is room for it. Stationary engines, locomotives, &c., are generally made horizontal, the exceptions being when room is of importance.

In petrol motors there is little choice between making the motor vertical or horizontal, but the horizontal engine seems to allow of the best shaped combustion chamber in some ways. The difficulty, as a rule, has been to make a really good arrangement of the various parts and at the same time allow of the engine being easily accessible.

The ideal arrangement, if one could make the details satisfactory, would be to have the engine and all the gear in one piece in the middle of the car, and to have the seating accommodation as nearly as possible between the wheels. There is a close approximation to this in some types, and the cars of some of these are being sold in considerable numbers.

American Type of Light Car.—The first of these arrangements is

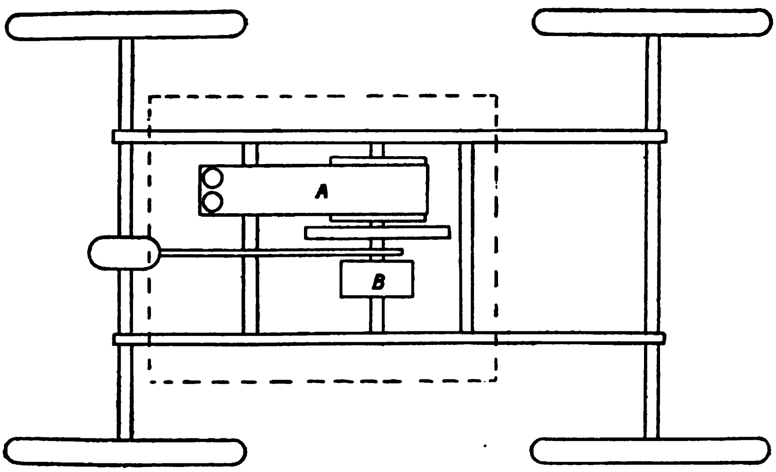


Fig. 321.
A, Engine. B, Change-speed gear.
Dotted lines show approximate position of body.

shown, in diagram, in fig. 321. It is a type, of which large numbers have been made in the United States for running in towns. The engine and the crank shaft are placed well in the middle of the car and the cylinder behind it. The position is, in fact, that of the old Benz engine reversed. The drive is a direct one to the back axle with a chain, and there are generally only two speeds worked by a Crypto gear.

This design has been varied by using two cylinders opposed to each other.

This type has had a very large sale, generally in the form of very cheap runabouts with single-cylinder, slow-running engines of large size for their power. It is a convenient type in some ways as long as it is only built for two passengers, but when carrying more than two it is not so easy to get at the various parts. It is also necessary to have the engine somewhat high up in the car to allow of its being reached above the back axle.

Duryea Car.—A variation of this type, which has many good points, is

the Duryea (fig. 322). In this a three-cylinder engine is used, which is of ample power to take the car up all ordinary hills on its top speed, and there are only two speeds. The car has no less than 15 cubic inches of cylinder capacity per cwt., and is, therefore, very near what has always been my

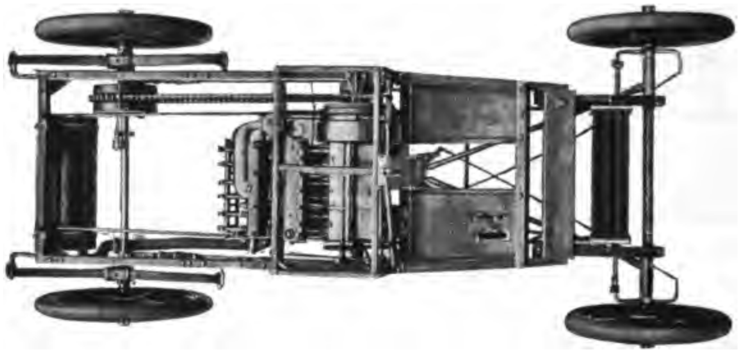


Fig. 322.

ideal: a light car with a powerful engine and only two speeds. The drive is direct to the back axle with a chain on the top speed.

Probably the craze for speed on the flat and also possibly the somewhat unusual appearance of the car have prevented this having as large a sale as it otherwise might have done.

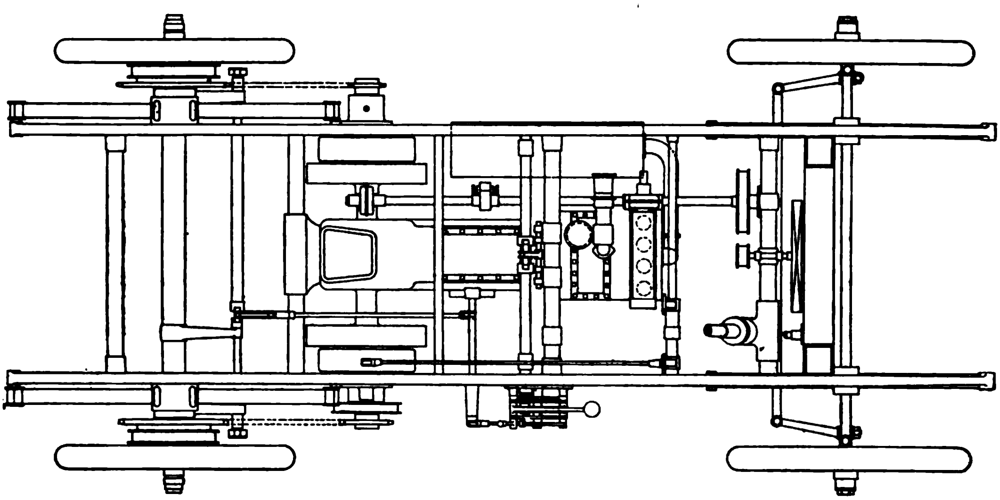


Fig. 323.

Singer Car.—Another type of car with horizontal engines is the Singer (fig. 323). This has the engines with the cylinders forward and long connecting-rods to the crank shaft, which is behind the centre of the car. In order to make the best of this design it seems that the engine should

have a very long stroke, somewhat after the style of a stationary engine, as the crank case has to be long enough for the long connecting-rods, and the extra weight consequent on lengthening the stroke is very small. This car has a direct drive with chains on the *two* top speeds, the low being by Crypto gear. Altogether there are many points in it well worth consideration.

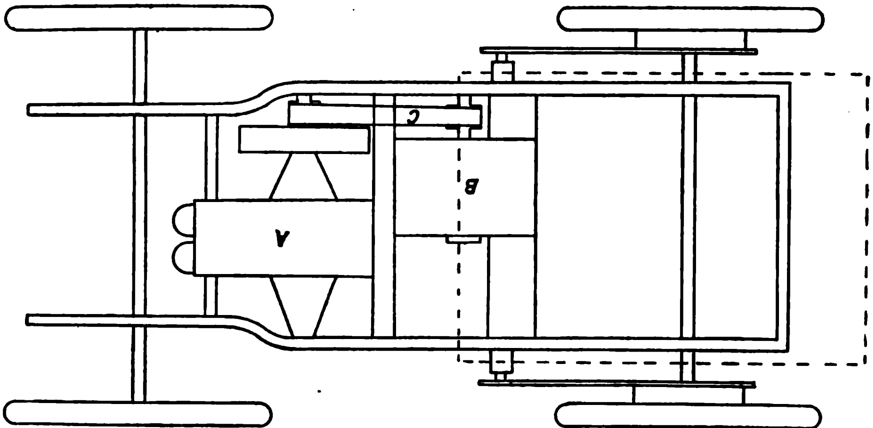


Fig. 324.

Wolsey Car.—A car which has had a very large sale is the Wolsey (fig. 324). In this the engine has the cylinders forward as in the last, but is of ordinary proportions, and drives the gear box with a chain, and the wheels from this by chains in the usual way. This admits of the weights being placed nearer the centre of the car than in the ordinary type of car;

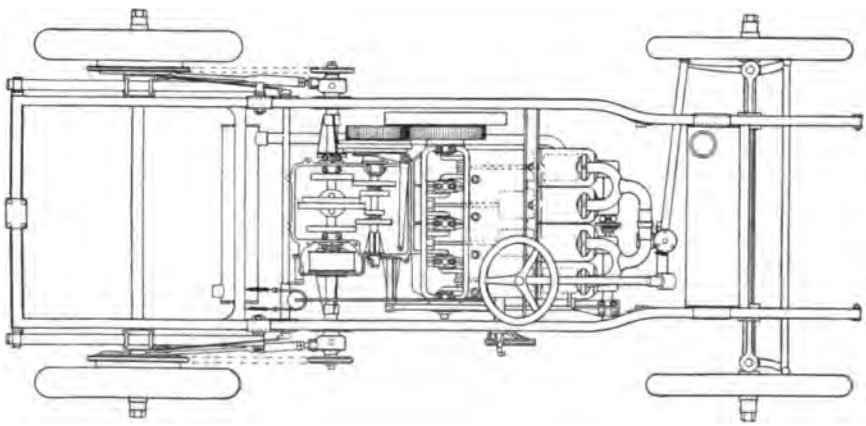


Fig. 325.

but, with a short-stroke engine, not so much as might be desired. The multiplicity of chains is also an undesirable feature, and the engine is not as accessible as it might be. The Wolsey Company are now making a car with a vertical engine of the usual pattern.

James & Browne Car.—A car with a somewhat similar arrangement

of parts, but with the above objections removed, is the James & Browne. This is shown in fig. 325. In this the engine and gear are so arranged that all the joints are machined, and there is no erecting of the gear box and engine in line on the frame as they are all practically in one piece. There is no chain between the engine and gear box, the drive being by gear, and the engine is very accessible, as all the parts that require attention come under the floor boards of the front seat. The weight is also well in the middle of the car. The final drive is by side chains in the usual way, and at present the cars are made with a shaft-to-shaft drive, but there seems no reason why they should not be made with a direct drive in the gear box, if desired.

Altogether this arrangement seems a very satisfactory one in every way, except that many people might prefer a car without the side chains. On the other hand, there is no bevel gear which is also objected to by some.

Fig. 326.

Fig. 326 shows this type of car as built with a side entrance body, and shows how much better it lends itself to this in many ways than the ordinary type. As the front axle is under the dashboard, the back wheels can be put well behind the body without excessive wheel base, and there is ample room for the door to the back seats without fouling the back mud-guard. Altogether this seems a very satisfactory design in every way and well worthy of study.

All the above cars have their crank shafts placed across the car, but there seems no reason why a very good design should not be made with the crank shaft longitudinal and the cylinders across. Several have been made in this manner with horizontal engines, generally of the opposed type, under a bonnet in the front of the car; for instance, the Wilson & Pilcher and Arrol-Johnston in England, and several makes in the United States, but these have no special advantage in distribution of weight over those with vertical engines which are generally more accessible.

One car, however, which has both the engine and change-speed gear well in the middle of the car is the Lanchester, shown in fig. 327, in the general

form it has had since 1901. In this the engine is of a special two-cylinder type with opposed cylinders, the heads of which come well out to the sides of the body and are easily accessible. The shaft is longitudinal and drives a live back axle with worm and wheel, the drive being direct on the top speed, and

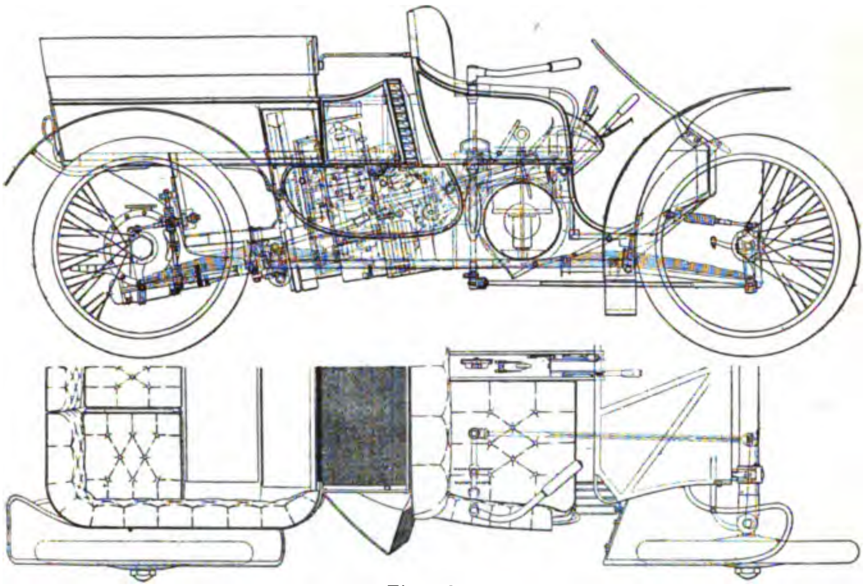


Fig. 327.

through Crypto gears on the lower ones. This car has a very great number of original features, among which may be mentioned low-tension magneto ignition with the magnets in the flywheel, automatically retarded when starting, special easy springing, tiller steering, surface carburettor, abso-

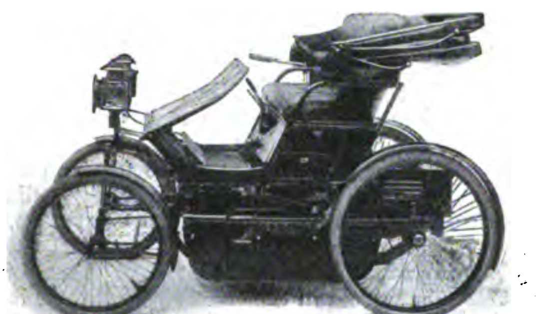


Fig. 328.

lutely balanced engine, tube front axle, dashboard thoroughly protecting driver, &c., too numerous to go into here, but in many cases anticipating their applications to foreign cars. Fig. 328 shows a very early form made in 1898 which embodied most of these features.

The form illustrated has been abandoned by the Lanchester Engine Co. in favour of one with a vertical engine placed between the front seats, which is shown in fig. 329. This car has also many special features, and the engine is so designed as to take up very little room, while, as will be seen, the removal of the bonnet makes a much more satisfactory design of car than the ordinary in many ways. In this car also the protection of the driver has been well considered, as the side doors to the front seats come up to the top of the dashboard, which is much better than only coming halfway up as usual.

Although, as mentioned, the type with the horizontal engine in the middle of the car has been abandoned by the Lanchester Engine Co., a large number are still running, and it is quite possible that it might be again worked out with a four-cylinder engine and made a very satisfactory design.

All the above types of car are worthy of careful study, as it is quite possible they may form the basis of the future car. Although in some cases the makers are giving up the special type and making cars with a vertical

Fig. 329.

engine in front, it does not necessarily follow that the principle of the latter is superior. In the first place, it is the business of commercial concerns to make money, and if they find that fashion makes one thing sell better than another they will make it regardless of the merits of the two. In the second place, machines which are excellent in principle often do not come into use, owing to defects in the details, while it is much more difficult to get the details right in any special design than in an ordinary one, as in the latter case the experience of others is a useful guide.

The light American type (fig. 321) must have some considerable merit or it would not have been sold in the enormous numbers it has been. One firm alone is advertising that they sold 4,000 last year. It is not so easy to make a nice design for a car on those lines with four cylinders.

Commercial Vehicles.—There is no doubt that there is an enormous field for commercial vehicles for the conveyance both of goods and passengers; nevertheless, the amount of these in actual use is comparatively small. Pleasure cars are to be seen in abundance, but a lorry is seldom met with. There can be no doubt whatever that the main reason of this is that lorries

have failed to pay. There is no sentiment in business, and if a thing pays better than another it will be used. There is no "prejudice" in the sense of those who like horses, while the taxes and other legal expenses special to the motor can have had but little influence in bringing about this result. The essential point is that most of the early attempts in this direction were not successful.

Steam was used for the majority of the early vehicles, and is largely used still. It is not necessary to refer to these, except so far as an indication of how to make (or not to make) internal combustion machines. It may be well to refer slightly to the advantages of each for a moment.

The advantages of the steam engine are that the experience with it has been long, that most mechanics are familiar with it, and that the engine can pull when not running, and consequently it can start without a friction clutch. There its advantages seem to end. It burns coal, it is true.

A great deal has been made of the "flexibility" of a steam engine. As a matter of fact, this has been much exaggerated. If an engine will give, say, 20 horse-power, it can be worked at any power between 0 and 20, whatever the motive agent may be. In many cases steam engines have been rated much below their power, and then have been run up to their full power to show their "flexibility." This is misleading.

The real flexibility of the steam engine is that it will pull well when the engine is moving slowly or is standing still. When the speed of a motor is reduced below a certain point it ceases to pull well, and if the load is not taken off it stops. Then it has to be disconnected from the load before it can be restarted. On the other hand, the steam engine will start itself under load, and generally does so. It has from this been assumed that change-speed gears can be dispensed with when using steam, but this seems a mistake. All traction engines and most steam lorries have a two-speed gear, as experience has shown it to be necessary. A very prominent steam pleasure car has recently had a two-speed gear added to it as an improvement. The fact is that the range at which a steam engine, particularly of the compound type, can run *economically* is very small.

On the other hand, the oil motor is far lighter per horse-power, and much more compact. In fact, if a petrol engine has the same weight as the ordinary steam engine and boiler on a lorry, two speeds would suffice, and the running would be better than when steam is used. Generally, the preference is for more than two speeds and a smaller engine, but the wisdom of this is doubtful.

As steam lorries have been tried for a long time without coming into general use, it may be well to look at the causes of the early failures. It cannot be simply due to their using steam, as this is successful for all other kinds of haulage, and can do all kinds of work more economically than horses. It seems to have been due to so many lorries having been built without due consideration of the work they have to do and to the consequent high expenditure required for repairs. In all other kinds of work the machines are the result of long experience, and are made of such proportions as not to require excessive repair. It was assumed that the sizes found suitable for a motor car would be equally suitable for a lorry, no account being taken of the difference in working conditions, the lorry being used for six days a week and nine hours a day, while a car is worked for only an hour or two a day, and not every day. Again, a lorry is intended to carry its full load under all circumstances, while the full power of a

car is rarely utilised. Consequently, parts suitable for a car are not strong and durable enough for a lorry. Moreover, most cars have required an amount of repairing which would be intolerable in a vehicle used for commercial purposes.

From all this it will be seen that car practice is not so likely to be of use as ordinary engineering practice. This is what the pioneer builders of steam lorries have found, and they have gradually abandoned car practice for what is a near approach to traction engine practice. Axles, gear wheels, &c., which were first made like those in cars, have gradually been enlarged to the sizes usual in traction engines; and such traction engine makers as have taken to manufacturing light tractors or lorries are doing very well with them. In fact, some of the most successful of the heavy-goods work is done by regular traction engines made in conformity with the Motor Car Act, which have locomotive boilers, simple unenclosed engines on the back of the boilers running at a moderate speed, their competitors being the water-tube boilers and high-speed enclosed compound engines of the "motor" lorry builder.

The road to success, therefore, for heavy work is a thorough study of traction engine work rather than of motor car work.

The work of the commercial motor is mainly of a four-fold character, viz.:—(1) Light delivery work, such as is now done by one-horse trotting vans; (2) lorry work; (3) tractor work; and (4) omnibus work.

The competition will be keen between the tractor drawing its load on a separate truck, and the lorry carrying the load and motive agent on the same frame. Wherever practicable, the preference will be found no doubt to be for a tractor with the load on a separate truck. By doing this the weight on each axle will be very much lessened, and there will also be no difficulty in having wheels of any desirable size. This is a point that will be of great importance. Much of the difficulty with the present types of lorries would have been avoided if the wheels had been larger. The small wheels follow the unevenness of the ground more closely than large, and if this is rough the machine is subjected to injurious vibration. Traction-engine makers found out long ago that for a heavy load, such as 7 tons on the back axle, wheels from 6 to 7 feet were required to make a durable machine. Yet there are lorries carrying this weight on the back axle, the wheels of which are less than 4 feet in diameter. The lorry is much more speedy than the traction engine, and should, therefore, have larger wheels than the traction engine for carrying the same weight. With the ordinary arrangement of a platform over the wheels they can hardly be made of a reasonable size. The tractor has the advantage, also, that it need not be standing still while the waggon is being loaded, as there may be two waggons to one tractor. In most ways, therefore, the tractor and separate waggon seem better than the self-contained lorry. The latter sometimes has the advantage of handiness in being able to manœuvre within a very small space; but if the lorry has a waggon as a trailer, as is frequently the case, this advantage is lost. In fact, the best way of moving heavy goods in quantities on the road is to have heavy tractors and two or more waggons. This would undoubtedly have been the practice long ago if it had not been for obstructive legislation. At present, perfectly satisfactory steam-driven tractors, or traction engines, whichever you like to call them, are obtainable.

There are no special legal restriction as to oil lorries or tractors as compared with steam, and as the former is *in principle* the most convenient

it ought to displace the latter for all road work if equally well designed and constructed.

Another weak point in many of the attempts to work heavy lorries has been that the speed at which they have been driven has been destructive both to them and to the road, while there is no need for such rapid carriage of heavy goods.

The success of motor commercial vehicles will mainly depend on not attempting to do too much with them. For the carriage of heavy goods economy, as compared with horse-drawn goods, should be the first consideration and speed the second. This also applies to omnibuses and char-à-bancs, but here greater speed may be allowed as passengers will possibly pay higher fares for it, but even for them it will be important to have an engine governor, which cannot be put out of action by the driver, to keep the machine down to a moderate speed.

In other ways the designs may at present follow much on the general lines adopted for motor cars, but allowing greater margins of strength and wearing surface, or possibly a special type of design on quite different lines may be developed.

Light Vans.—The demand for an economical light motor van will probably be very large, and it should be easy to make, as the ordinary pleasure car, made stronger where greater durability is requisite, would probably give satisfaction. The main differences would be (1) much larger wheels, particularly the driving wheels; (2) greater reduction from the engine to the road wheels, as the speed should be slower and the wheels bigger than in the car. This will be to a certain extent compensated for by the fact that the engine should be run slower, and perhaps it may be found that it is best to utilise the same gear as the car and run the engine very slowly. Taking 4 feet 6 inch wheels, and 15 miles an hour, the gear ratio would be 5 to 1 at 500 revolutions and $7\frac{1}{2}$ to 1 at 750 revolutions. It would be possible to make a van with a live axle and 5 to 1 gear satisfactorily; but not a greater ratio, unless with a worm drive. With side chains a reduction of 7 or 8 to 1 and a direct drive in the gear box could quite well be arranged, but it is an open question whether side chains will stand well enough for commercial work. Fifteen miles an hour is assumed to be the extreme speed necessary for a van; it might well be less and the reduction be greater in accordance therewith. One plan that could be used would be shaft-to-shaft drive in the gear box with the top speed somewhat reduced. If the top speed was $1\frac{1}{2}$ to 1 with a 5 to 1 reduction at the back axle, the total reduction would be $7\frac{1}{2}$ to 1. Some such arrangement as this may be the best, the engine speed being limited to a maximum of about 750. In this class of van there is generally no difficulty in having in big wheels, as the platform does not come outside the wheels. Whether for this class of work ball or plain bearings are best remains to be seen, but it is quite certain that if balls are really the best for cars they are also so for vans. In fact, the gain on the vans will be much greater than in the cars, as they are worked longer hours, and, therefore, economy both in fuel and repairs is more important.

It remains to be seen if these vans will run on iron or rubber tyres. The present tendency is for rubber tyres exclusively. These are very nice in many ways, but are expensive. In many cases, where iron tyres have been used, it has been found that the vibration caused more wear and tear of the machinery than the cost of running on the rubber tyres. This is very likely the case, but, if so, a good deal of it could probably be avoided by

experience in the machinery. It will depend rather on the speed required for the van; and for most work a moderate speed and iron tyres may be the most economical.

Lorries will be adapted for speeds of about 6 miles an hour, and the reduction be more than 10 to 1, which would necessitate an extra gear drive, besides the ordinary one. Except for the engine and change-speed gear the whole machine is well within traction-engine practice, and the trade will probably gravitate into the hands of the traction-engine maker. The only essential difference is in the matter of allowing for the motion of the springs. This may be done by chains or universal joints on one of the shafts, or simply by a radius rod on the back axle, keeping the teeth in gear. The latter is the simplest and cheapest plan, and seems to work quite well. There seems no possibility of anything but iron tyres being used for this class of work, so, to avoid vibration, the wheels must be big and the speed moderate. There has, as yet, been little development in motor tractors, but it will probably become considerable. The best arrangement has yet to be determined, but it might be much the same as in steam tractors—that is, say, a horizontal engine over an oil tank, with the ordinary traction-engine gearing, or a vertical engine with bevel drive gear, as in the motor car pattern.

It is very doubtful if the Ackerman steering arrangement is as good as the old pivot for lorries and tractors. The pivot is still in universal use for traction engines, and has the great advantage that, with it, the front wheels are more easily turned when the engine is standing still. This is often necessary when manœuvring in a small space. It is also simpler and cheaper.

Omnibuses are a rather difficult problem, as there must be a considerable reduction of gear, and the body must be very near the ground. The wheels must be large enough to wear well. At present most of the 'buses running have very small wheels, and it is possible that under their usual conditions—viz., running on paved roads with rubber tyres—these will be successful, but larger wheels might be a great improvement. The difficulty here is the use of any kind of live axle which requires the gear wheels to be near the centre of the car. Possibly, for this class of work, a dead axle cranked under the body of the 'bus would give the best results. Rubber tyres are considered to be necessary when the speed is great. A permanent type can only be decided on after much consideration and experience.

Motor char-à-bancs should be in great demand, as they are easier to make than 'buses, can make longer and quicker journeys than horse-drawn vehicles, and have an extensive choice of routes—advantages which should secure good fares. As the whole platform is high up there is no restriction as regards the size of the wheels or the kind of axle that can be used.

There are several other possible uses for motors that have hardly been touched yet. All sorts of agricultural work might be advantageously done by them. Ploughing, in particular, offers a wide field. The main objection to steam ploughing is the enormous consumption of water, all of which has to be led. This forms a very large part of the cost; if this cost could be dispensed with mechanical ploughing would soon pay. A motor plough would only burn something like a quarter of the fuel of a steam one, and this would save a good deal of leading. To make the thing a real success it will be necessary to arrange for *all* the work on the farm to be done by motors. It is no use substituting motors for horses in one thing if horses

are employed for other purposes, and often kept idle. Motors are especially serviceable for fire engines, as they can be started more promptly, pump more water for a given weight, and cost less for a given power than steam engines. Engines of this kind are, of course, self-propelling, and the antiquated plan of getting horses to draw them to the fire is avoided. Often they can reach the fire in less time than the horses for the present type of engine can be harnessed.

Portable centrifugal pumps are often required for various work, and could very well be direct coupled to motors. In sinking foundations the whole motor and pump could often be lowered down where wanted, when it would be quite impossible to get an engine and boiler there.

For electric lighting there should be a good field for direct-coupled motors and dynamos.

There will be many more uses that may be suggested, but the above, as samples, show that the possibilities of the motor of the future are enormous.

Marine motors will be one of the most important, but have been dealt with.

CHAPTER XXII.

RACING CARS.

A VERY brief consideration of the question of factor of safety and wear will show that the construction of racing cars is a very different matter from that of touring cars. This is, of course, the case with other things, as, for instance, boats. In a racing machine of any kind the main point is to obtain the greatest possible power for a given weight and sufficient durability for the occasion. This idea has been carried to an absurd extent in racing yachts, some of which have cost £50,000; and, after taking part in half-a-dozen races, were broken up because they were utterly unfit even for pleasure sailing in fine weather. Consequently, all yacht-racing rules are made on the assumption that building a yacht for the purpose of winning a race is bound to deteriorate her useful qualities, and all that is hoped for is to make rules which will keep this deterioration within such bounds that she is not absolutely useless at the end of her racing career. It has been found necessary to have very elaborate rules for this purpose, and the most successful racing classes are those in which all boats must be built exactly alike, the winning of the races depending on the skill of the crew, and not on the boat.

Motor racing has not been practised long enough for the development of such special machines, but, if continued long enough, every useful quality will be sacrificed in order to get speed, as has been the case with yachts.

A consideration of the construction of racing cars in all its bearings would require a volume, and all that is called for in this work is a general reference to a few points only.

In present day racing, in the large open races, the only restrictive rule concerns the weight. Under these circumstances the main consideration for speed is to develop the greatest power with a given weight. Racing engines can be built with a limit as low as about 3 or 4 lbs. per horse-power; hence every pound saved means the gain of a quarter of a horse-power. As the difference between winning and losing may be a very small amount of horse-power it is evident that every pound difference in weight is of the greatest importance.

The main reduction in weight is made in two ways. In the first place comfort, quietness, &c., are of no importance, and everything conducive to these can be done away with. The body need only consist of two very small seats, one for the driver and one for the mechanic, the latter being practically on the floor, and something for them to put their feet against instead of floor boards. The exhaust pipes need only be long enough to take the smoke away from the driver, and there need be no silencer. The seats have also been arranged sometimes to form tanks for either petrol or water, or both. As an instance of how far this leaving out of important parts may be carried the side brake in some racing cars has no quadrant for keeping it on, the saving in weight being about a couple of pounds.

The other way is by the reduction of the factor of safety and durability.

It is obvious that if the part is to last a short time before breaking it may be far lighter than if it is to be permanent, and the weight may be lessened still more if there is no objection to the chance of a certain percentage of breakages for the time required, and a less reliable material of higher initial strength can be used. The limit to cutting down the factor of safety depends solely on the recklessness of the maker and driver of the car.

In addition to these there are many ways in detail in which weight may be reduced. Taking the engine first, the main consideration is that we must run it as fast as ever it can be got to run. This, of course, means a great deal of attention to the weight of the moving parts, which must be made as light as possible, regardless of cost or durability. Pistons machined out of steel forgings can probably be made lighter than anything, and connecting-rods could be made lighter out of forgings than stampings, as stronger qualities of steel can be used. The valves on the top of the cylinder, either inclined or vertical, make the best shaped combustion chamber, and so seem the right thing. High compression is probably essential. As the engine is not required to run slow, but fast, the flywheel can be cut down to a size just sufficient for starting the engine. The older construction of crank case (fig. 154) is probably lighter than that shown in fig. 155; if so, it is preferable, as, of course, nothing will be done to it during the race, while the cost of tuning-up before or after is no drawback. The smallest bearings possible will be used that will run cool with the most copious lubrication. Unfortunately, water cannot be used to cool them in a car, as in a boat, in which heated small bearings are cooled by turning a hose on them.

A slight saving of weight might be made by carrying all the bearings on the top half of the crank case, and making the bottom half simply a very light sheet-brass casing.

The cylinders are the great weight of the engine, which it is difficult to reduce. Steel cylinders do not seem to have been entirely successful, probably because the tops have always been cast, and, further, because the cast-iron cylinders stiffen the crank case. If the valves are at the top a small amount of weight can be saved by making them with the seatings cast in the cylinder, instead of loose. This entails taking off the cylinder when a new valve has to be supplied, but new valves put in just before a race might be calculated not to require renewal during the race. There seems no definite reason why cylinders should not be made out of a solid block of steel, the inside being machined and the outside machined as far as possible, and then filed and chipped down to the required thickness. It would be very expensive, but it should make a very light cylinder.

The diagonal or opposed engine has the great advantage of the crank shaft and crank chamber being short for the number of cylinders, and therefore should make the lightest engine. The difficulty is that the shaft cannot be balanced well and the impulses divided evenly unless eight cylinders are used, and it is not certain that an eight-cylindered engine is lighter for its maintained power than one with fewer and larger cylinders.

It is possible that more power might be got by some form of scavenging, such as is shown in fig. 212. So far, reducing the size of the cylinders seems to have been a more successful way of making them light than special means of construction.

In the car the principle of reducing the size as far as possible may be

adopted, and the modern racing cars are of relatively short wheel base. Also, the narrower they are, the lighter can the axles, &c., be made, and this depends on making the centre of gravity as low as possible. Springs may be cut down in weight considerably, as they need not be long enough for comfort, but only just long enough to keep the wheels on the ground.

In general construction little has been done to alter the general design of touring car. One very successful car last year, however, had two alterations which may be a forecast of further change. It had no gear box and no differential, the change-speed gear being on the back axle, which was a solid bar, not in two pieces. The absence of the differential seems at first sight likely to involve great wear on the tyres, and in a touring car would do so; but possibly, in a racing car, it actually involves *less* wear. This is because, at the speed a racing car runs, it will have little weight on the inside wheels going round corners. If there is a differential, this wheel will therefore slip under the driving stress, possibly to a great extent. If both wheels are solid with the axle it will, of course, only slip to the extent of the difference of speed in the two wheels, and the car will get the full benefit of the engine power.

By having no differential, it is possible to arrange the change-speed gear on the back axle without undue complication. The weight need not be great, as, of course, the gear wheels need only be strong enough to stand through the race, and need not have any durability.

It seems that if we do away with the differential we might also do away with the whole of the axle casing, and save a great deal of weight that way. In the ordinary live axle the casing takes the whole of the stress, but, if the axle is continuous, it does not do so, and many lorries and all traction engines have no casing to their axles. The saving in weight, in this case, would be considerable.

One point, in both engine and car construction, of importance in racing is that a reduction in weight is made by using bolts and studs with finer pitch threads than is usual. Whitworth threads, for instance, are sufficiently coarse to stand screwing up and unscrewing many times, and yet leave ample margin for holding. The thread, however, cuts a long way into the bolt, and this weakens it a good deal. By screwing it with finer threads the amount cut away is less, and smaller bolts can be used, but the threads must fit well, and, if very fine, get useless after being screwed on and off a few times. This would not matter in a racing car, as they could be frequently replaced. Bolts are also made stronger by having a hole drilled up them as far as the beginning of the thread, so as to reduce the section of the bolt to that of the bottom of the thread, and are at the same time slightly lightened.

The above remarks are generally *apropos* of cars built under no limitation except the weight. The same general principles will, however, apply to any other rules. That is to say, everything should be left out which does not come under the rule limits and everything done that will increase power consistently with the rule. Thus, if petrol consumption is the standard, everything must be regulated accordingly. It will probably mean a single, or, at most, a two-cylinder engine, running as slow as possible, and, in fact, approaching as near as possible to the stationary gas engine. Every other weight must be cut as fine as possible to get the best result in this. It will, no doubt, have a cut-out governor, and probably a hand control to the carburettor. It will also want as great a range of speeds as

possible, in order that the engine may be run with the slightest variation of speed.

So with any other limitation. This is well recognised in yacht racing, where it is always assumed that the type of boat produced by any rule will be an exaggeration of all the permitted points.

TRIALS.

The progress of the automobile movement can, to a great extent, be summarised by the results of the various trials and races which have taken place. Of these, by far the most instructive are the various reliability trials which have at different times been held by the Automobile Club of Great Britain and the Scottish Automobile Club, and particulars of these are given in Tables i. to vi.

In all these tables the cars are arranged in the order of the actual mean speed in the various hill-climbing tests made during the trial in which they were engaged. As a rule, no particulars are given of any cars which failed to complete the trials and to get up all the hills in the hill-climbing trials without shedding any part of their load.

The petrol consumption is given in car miles per gallon, as this is the ordinary expression for it. It is also given in passenger miles per gallon, which reduces the cars carrying different numbers of passengers to a level; also in ton miles per gallon. The latter is really of only scientific interest, as the information a car owner wants is the amount of petrol he will use for a certain amount of work done, and weight is of no advantage.

The actual weights and cylinder dimensions are given as far as can be ascertained, but in some of the earlier trials it does not appear that the cars were always very accurately weighed, makers' weight occasionally being taken.

The cylinder capacity per cwt. is given, as this is in many ways a test of the success of the designer in putting large power into a light weight. No calculations are made based either on the stated horse-power of the engine or the stated revolutions at which it runs, as these appear to be matters which are governed more by commercial considerations than by actual performance. In practice most makers run their engines above what is stated to be the number of revolutions for their normal and often get more power out of them than they are professed to yield. The extent to which this is done varies with different makers from purely commercial reasons, and, therefore, such calculations would be of no technical value.

If the data are sufficiently reliable an estimate is made of the power actually produced at the road wheels and also of the power per 100 cubic inches of cylinder capacity. The two estimates together give the efficiency of the engine in producing power and also the efficiency of the gear.

Before considering the development of the motor car, as shown by these figures, we may glance at the effect of different formulæ used for hill-climbs and trials. It may be taken for granted that makers taking part in these competitions will study the formula under which they are to be marked, and will endeavour to construct vehicles which will perform well under it. In fact, from a commercial point of view this is a most important thing to do.

In races the matter is perfectly simple. Speed is aimed at and nothing else. Certain limitations are introduced, but the effect of racing will obviously be to foster speed at the expense of all other useful qualities, as

far as is consistent with the rules. Thus a weight limit will exclude cars of unreasonable weight and a petrol limit will keep the consumption within bounds, but any designer of racing cars will naturally develop speed at the expense of other qualities as far as he is allowed.

Reliability trials, on the other hand, test those qualities which are useful to the buyer, and in all these some system of marking the cars is adopted, the object of which is to assign the highest marks to the car which will be most generally useful. This is a very difficult thing indeed to do perfectly, as many of the qualities which are most needed are very difficult to judge in a comparatively short trial. The trials as a whole, however, have had the most excellent results, and have contributed very largely to the development of the motor car.

In regard to the question of reliability, the usual system has been to give a certain number of marks for reliability and to deduct one mark for every minute that the car was stopped on the road for any purpose whatever. This seems excellent in every way. The only question which arises is what position should be taken with regard to tyre troubles, as these are, in a way, matters beyond the control of the designer of the car and are largely a matter of luck. Tyres are usually supplied to the customer's choice, and are in any case not made by the car builder. While this is true, it is also true that the designer of the car has a great control over the amount of tyre troubles, as these depend mainly on the weight of the car and the size of tyres with which it is provided.

As price is generally in some way considered, either in the classification or marking, it is obvious that if tyre troubles were not considered at all there would be a premium on fitting cars with tyres not large enough to give good results in long continued use.

Further, it is well known that the thinner pneumatic tyres are the easier they run, and this would also help in the hill-climbing and petrol consumption tests. For this reason it seems best that makers should be responsible for their tyres and choose what they consider will give the best results, though there are great points in keeping the records of tyre failures separate from others and marking them differently.

That the tendency to fit insufficient tyres is a real one is shown by the fact that in the Thousand Mile Trial of 1903 the smaller and cheaper cars had far more trouble from their tyres than the larger and heavier ones. This was obviously due to the wheels and tyres being made too small for the work they had to do, in order to save expense.

More difficult points to deal with are petrol consumption, weight, silence, &c. Of these the most difficult to deal with is probably weight.

If weight is neglected altogether it is evident that the lightest car has the greatest advantage in hill-climbing, petrol consumption, and every other way; yet it is certainly undesirable to reduce the weight of a car so that it is unfit for long continued use. On the other hand, some of the systems of marking, both for long trials and hill-climbing tests, have put a premium on weight which is out of proportion to its merits. It will, in fact, always be the most important thing to have a car as light as possible consistent with durability, as the most important point in upkeep is the tyre bill; and this is almost entirely dependent on the weight. In working out technical results it is necessary to take it into account in order to get special information, as, for instance, the ton miles run per gallon of petrol; also the horsepower yielded by the engine on hills. To give marks on this basis, however,

puts a very distinct premium on weight, as it allows of the petrol consumption being in proportion to the weight, also of the speed up hill being reduced in proportion as the weight is increased. Thus, if one car weighs 24 cwts., and another 36 cwts., the latter may burn 50 per cent. more petrol and go 33 per cent. slower up hill, and still get equal marks, even though it only carried the same number of passengers. Let us assume that the two cars above had the same engine in them, and, generally speaking, the same arrangements, but that the heavier car was geared lower in proportion to its weight. It is probable that on a system of marking which put such a premium on weight the heavier car would win, as it seems that the resistance increases more slowly than the weight, but more rapidly than the speed, on account of the extra air resistance. Hence, under such a formula, it would pay either to build a specially heavy car or to carry ballast. Even if this is not done it puts heavy constructions on an equality with those which, by superior design and construction, are lighter.

In the matter of petrol consumption there is nothing to note, except that the most economical adjustment of carburettor, &c., does not always result in smooth running, and that, on the whole, the fewer cylinders there are the more economical a car ought to be. Consequently, there will be a tendency to tune up cars specially and increase the compression, &c., for petrol consumption trials, but this can hardly be avoided. If a good number of marks are given for silence and smoothness this will be checked.

There is one other point in which the systems of marking cars may have bad effects, and that is in the matter of hill-climbs. In these the main factor taken is usually the speed up the hill. This may, in some cases, have a tendency to induce makers to overgear their cars on their bottom speed. It is obvious that a car will do its best speed on a hill if the gear in use on that hill is such that the engine will just run the revolutions at which it does its best power. This means, generally, that the engine is exerting very nearly the maximum torque it is capable of. That is to say, that its gear is such that a very small increase of resistance or a very slight falling off in power would prevent the car getting up the hill at all. If the hill on which the trials are made is known to be the worst that the car will ever be wanted to go up this will give a slight margin, but not enough to be quite certain of not having to shed passengers in ordinary practice, as the cars run at hill-climbs are naturally tuned up to do better than they can be relied on to do in daily life. Also, hill-climbs take place mainly in the summer when the roads are in good condition. It is much better to have the gear lower than this, as it is far better in ordinary life to get up a hill slowly than not to get up it at all, but a car so geared would be at a great disadvantage in speed competition as compared with a higher geared car.

In some tests cars have been made to stop on the steepest part of the hill and re-start without running back more than a certain distance, say, a yard or two, this being included in the time; and this is an excellent plan, as an overgeared car will not be able to pick up quickly enough to do good time. If this is done on a hill with an inclination of, say, 1 in 6, with a full load of passengers, it will be pretty certain that the cars which win are geared low enough for all ordinary purposes.

Meanwhile, it should be remembered that it is perfectly possible to make any car go up any hill on which the wheels will grip, and that this does not need a powerful car, but only that the low-speed gear should be properly droportioned for the work it has to do. The matter is of such importance

from a user's point of view that it seems as if it would be an improvement in future trials to either absolutely disqualify any competing car which failed to get up any of the hills on the route with its full load, or, at all events, to penalise it by a really large number of marks, so as absolutely to prevent its taking a high place compared with cars which do not fail. It is for this reason that in the majority of the tables of results cars which failed to take all their passengers up all the hills are omitted.

For these reasons the marks gained in the various trials are not given, more especially as they were on a different system at each trial. The complete reports of the various trials will well repay careful study, but only a certain number of points can be dealt with here.

The first trial taken is the one at the Richmond Show of 1899. In this we can see the germs of most of the points of the modern car, but the points are scattered about in separate cars. The general arrangement of the modern car as regards engine position, change-speed gear, clutch, &c., was found in the Daimler & Panhard cars; but these had tube ignition, cut-out governors, shaft-to-shaft drive, and comparatively small cylinder capacity in proportion to the weight. The Panhard, it is true, had as much as 7 cubic inches per cwt., but the weight was taken with only two people in it; and with a body for four people, fully loaded, it would have been much less. Several makers, however, had cylinder capacities approaching the modern cars, the Benz cars having from 7·7 to over 9, and the Ducroiset 8·2, the latter including six passengers. These were, however, ignited with the old-fashioned low-speed trembling coil, and in other ways the engines were not adapted for running fast; hence the power in proportion to the size of the cylinders was very small. The Lanchester was the only car with more than 10 cubic inches of cylinder capacity per cwt., having 12·4, and being therefore, in this respect, fully up to most modern cars. The Lanchester car was also the only car with a live axle, and the only one with a direct drive on its top speed. The Lanchester and the Bergman were fitted with low-tension magneto ignition. The Critchley was the only car in which all the wheels were of the same size. The Barrière tricycle was the only instance of single-contact high-tension ignition.

Hence most of the points of a modern car were there, but not combined in any one car. The performance was of corresponding character, as shown by comparing these points with cars in later trials. Trials were held in 1900 and 1901, but the results are not now of very great interest from a technical point of view. In 1902, however, we come to a trial when cars were getting into their present form, which is of considerable interest; and in 1903 one which is of very great interest indeed. The latter is the first trial of any great duration in which the petrol consumption was measured for the whole trial, and, in fact, the whole trial was conducted much more carefully than the earlier ones, and the weights, &c., were more accurately taken. By this time, also, the majority of cars had attained their present general outline, and belt-driven cars, &c., were pretty well extinct. In fact, this really forms a starting point for comparison with all modern trials. In 1904 this was succeeded by a trial for light cars not exceeding £200 in value, and in 1905 and 1906 trials were made in Scotland.

It is interesting to compare the results of these trials. Taking the constructional points first, the most important one is the ratio of cylinder capacity per cwt., and the power yielded by the cylinders. The maximum cubic inches of cylinder capacity per cwt. were :—

1899— 8	H.P.	Lanchester,	12·4
1902—20	„	Pascal,	11·5
1903—18	„	James & Browne and 20 Humber,	11·1
1904— 9	„	Oldsmobile,	6·7
1905—24	„	Germain,	11·7
1906—35	„	Ariel,	13·2

The best power actually developed at the road wheels per 100 cubic inches of cylinder capacity was as follows:—

1899—	Barriere tricycle,	7·3
1902—	6 H.P. De Dion,	7·9
1903—	12 „ De Dion,	9·5
1904—	6 „ De Dion,	11·7
1905—	8-10 „ Humber,	8·3

These figures are not exactly comparable in all ways, as all the trials should have taken place on one hill and have been run in the same way. Obviously a car will do its best horse-power on a hill of uniform grade such that the engine will just run at its best revolutions with some particular speed in gear. As a matter of fact, the hills on which the cars were tried were usually of very varying grade, and in this case the engine cannot be running at its most effective torque and revolutions all the time. Further, in the 1903 and 1904 trials there were several hill-climbs, and the horse-power taken is that of the *best* hill. On the other hand, in 1899 there was only one hill-climb, and the vehicle which did the best horse-power for its cylinder capacity was only fitted with one speed. The more speeds a car is fitted with, and the nearer they are together, the better is the result that can be got out of the engine. In 1905 the power taken is the mean of two hill-climbs, one of them of great length.

In 1899 the best result was obtained from an air-cooled tricycle engine, and the next best from the Mors car with 4·5. The tricycle engine obviously would have given more power if water-cooled. In the next three trials the best results were from the De Dion engines, which were essentially the tricycle engines water-cooled. By 1905 the car makers had adopted the essential points of the tricycle engine—*i.e.*, their engines were run at high revolutions and had electric ignition suitable for this, as also light moving parts, large valves, &c. We see, therefore, that the increase in speed in modern cars has been principally due to the adoption of principles of engine construction which were thoroughly understood and developed by the tricycle builders in 1899, and not by an increase in the size of the engines in proportion to the weight of the car.

Turning now to petrol consumption, it is again difficult to make absolute comparisons, as the routes of the trials were different, and this would necessarily affect the consumption. Again, in the early trials, the distance travelled was small, and therefore any slight error there might be in the measurement of the petrol would affect the result a great deal, while the condition of a short bit of road may be very different from that of a longer one. On the other hand, petrol consumption is largely a matter of driving and tuning up, and this has developed enormously during the last few years. In the early days the marks given for petrol consumption were comparatively few, if any, while most of the drivers were sufficiently occupied in getting their cars to go through the trial at all without troubling very much about consumption; but later far more attention was paid to this.

The actual best recorded rates of consumption were as follows, in miles run per gallon of petrol:—

1899—Benz Ideal and Mors,	40
1902— 5 H.P. Peugeot,	36
1903— 5 „ Peugeot,	39
1904— 7 „ Swift,	39
1905— 8 „ Darracq,	37
1906—10 „ Darracq,	40

As some of the cars carried only two passengers, and some four or more, this does not give a very fair comparison between them, and, therefore, the more just comparison, from a purchaser's point of view, is the number of passenger miles per gallon. The best results for this are as follows:—

1899— 8 H.P. Mors,	160
1902—12 „ Century,	99
1903— 8 „ M.M.C.,	131
1904—Alldays,	103
1905—12 H.P. Arrol-Johnston,	125
1906—10 „ Darracq,	152

Neither of the above returns takes any account of the weight of the car. To do this we may express the consumption in ton-miles per gallon. The best results in this case are as follows:—

1899—10 H.P. Daimler Lorry,	47
1902—15 „ Panhard,	29
1903—12 „ New Orleans,	39½
1904— 6 „ Siddeley,	27
1905—12 „ Arrol-Johnston,	44
1906—10 „ Darracq,	42

In the 1899 trials the Mors dogcart and the Daimler phaeton both ran over 40 ton-miles per gallon.

Taking it all round the economy in petrol in the modern car is not very marked, though the results are naturally much more consistent. Several of the cars in the 1899 trial only ran about 10 ton-miles per gallon.

In the matter of reliability there has been an enormous gain. In the trial of 1899 the distance run was only 50 miles on an extremely easy bit of road, yet there were numerous stoppages in many of the cars. It is very difficult to give any statistics which will show the relative number of these in later trials, as the conditions varied a good deal.

The following figures give the actual number of cars which entered, finished, broke down, were disqualified, &c.:—

Date,	1899	1902	1903	1904	1905	1906
Entries,	25	80	135	38	44	84
Starters,	...	56	97	34	43	80
Non-stop runs,	8	1	5	4	15	24
Finished,	18	44	66	25	38	67
Broke down,	...	11	29	9	7	12
Disqualified,	...	1	2	...	...	1
Length of trial miles,	50	650	1032	600	595	671

Besides this there has been an enormous reduction in the number of small stoppages in cars which have actually finished without breaking down. This is largely due to better details in such matters as wiring, &c.

In considering modern cars it is in many ways best to take the 1903 trial as our starting point, though there is much to be learnt from a careful study of the earlier trials. Up to 1902, however, the voiturette and the heavy car had developed on very different lines, while by 1903 they were approaching each other so nearly as to be practically merged into one type. For instance, in the trials of 1901 the weight of the four-seated cars varied from 10¾ cwts. in the New Orleans and M.M.C. to 31 cwts. in the Napier. Eleven were under 14 cwts., nine were from 14 to 21 cwts., and nine over 21 cwts.

The heaviest four-seated car was just over three times as heavy as the lightest. As the side entrance body was not then in general use, the weight of some of the heavier cars could not be accounted for by the size and weight of their bodies and was due to heavy construction, while in the case of the very lightest the weight of some parts was unduly cut down so that they needed excessive attention. The mean of these was, however, to a great extent represented by the 7-horse Panhard and some others which carried four passengers on a weight of about 15 cwts., and have proved themselves thoroughly durable. By 1903 the majority of the vehicles had approximated to this type; the lightest four-seated car being 11½ and the heaviest 26½ cwts. Only four cars are under 14 cwts. and half of them come between this and 21, the majority of the rest being a little over 21 cwts. In 1901 all the four-cylinder cars were very heavy cars, the light ones having only one or two cylinders; but by 1903 there was little difference in weight between the four-cylinder cars and those with fewer cylinders, two four-cylinder cars being under 16 cwts. and many very little over this. These two cars do not appear in the table, as one of them was disqualified and the other failed to take all its load up one of the hills; but both makes have been practically very successful.

In comparing the results of modern trials we may profitably take the question of reliability first, as it is of the greatest importance. Beginning with the actual breakdowns which caused the cars to be withdrawn from the trials altogether, the reasons are as follows:—

1903 Trial.—In the 6-horse Eagle, 6½ Vulcan, 8 Elswick, 25 Maudslay, and Chenard & Walcker, no reason is given for withdrawal. The 10 Peugeot and 12 Clement were disqualified.

The others failed for the following reasons:—

3½ Rex. Clutch trouble.
5 Humber. Broken gudgeon pin and connecting-rod.
Pony Richard. Engine, hot.
6 Relyante. Burst tube to cooler.
8 Regal. Gear trouble.
10 George Richard. Accumulator ran out.
10 Rex. Differential gear.
5 Roots. Chain and exhaust valves.
14 Argyll. Broken connecting-rod.
10 Simms. Half-speed shaft gear wheel broke.
12 Krupkar. Broken cardan shaft.
15 Bellsize. Broken connecting-rod.
14 Brooke. Split pin fell into 2-to-1 gear and broke the teeth.
18 Star. Sprocket sheered from driving wheel.
20 Winton. Broke crank shaft.
16 Lanchester. Engine heated.
20 Spyker. Gear out of order.
12 Humber. Stripped driving pinion.
12 Wilson & Pilcher. Commutator and pump shaft gear wheel adrift.
18 Mors and 12 Ariel. Collison.
20 Holcar. Engine trouble and punctures.

This list shows some failures of important parts, apparently due to weakness; while a good many others are failures from bad details, such as split pins. The broken connecting-rods were said, in some cases, to be due to bad lubrication. There were, however, several failures of gears, differentials, &c., which were due apparently to the parts not being strong enough.

If we turn to the 1905 trial, we find the following complete failures:—

12 Darracq. Timing wheel fired.
18 Argyll. Cardan shaft bent owing to fouling a culvert.
15 Darracq. Clutch hot.
20 Drummond. Steering gear broke.
12 St. Vincent. Clutch trouble.
20 Ford and 30 Martini. No reason given.

In 1906 the failures were:—

6 Rover. Broken steering-rod.
14 Scout. Magneto trouble.
12 Victoria. Broken pinion.
18 Courier. Broken torque-rod to back axle.
18 Speedwell. Stripped differential.
18 Arrol-Johnston. Sand in engine chamber.
20 Maudslay. Disqualified.
20 Enfield. Broken petrol pipe.
30 Chenard & Walcker. Broken gearing to back axle.
25 Brooke. Contact on magneto failed.
20 Horbick. Spokes of shaft broke.
30 Peugeot. Clutch trouble.
24 Bellsize. Strained change-speed lever.

Some of these may have been due to small causes or bad driving, but the majority of them are serious breakdowns which should hardly occur on such a short run as 670 miles. At the same time, it must be remembered, in all these trials that a certain amount of new makes of cars are entered by new comers to the car trade, which are entered to get them a start in the trade.

There were two cases of broken pinion and one of stripped differential, which show that the margin given to these parts is sometimes much too small. Two magnetos failed and, in addition to this, one failed in a car which finished with accumulators.

Even in cases where a car succeeded in finishing the trial there were in some cases stoppages which, in the ordinary users' hands, would have entailed the abandonment of the journey, and, therefore, should be put in a different category to stoppages for adjustments of an ordinary nature. Cars entered for a reliability trial can be driven by very skilled mechanics and may carry a much more complete set of spare parts and tools than should be needed in ordinary use, and, hence, a car very seriously broken down may be doctored up so as to continue the run if enough time is allowed. It might be a matter for consideration whether in future trials a car which fails so seriously that, even with the most skilled attention, it cannot be restarted in reasonable time, say an hour, should not be disqualified on the grounds that a car which develops such defects in the few days a trial lasts is hardly fit for ordinary use.

In the 1903 trial the stoppages which would have come under this heading were as follows, the figures being the number of minutes the car was stopped:—

6 Regal. Ignition troubles, 112.
6 Elswick. Ignition troubles, 375 and 131.
8 Achilles. Ignition troubles, 89; exhaust valve, 127.
9 Beaufort. Ignition, 116.
9 James & Browne. Ignition and changing exhaust valve, 62.
12 Krupkar. Re-fitting exhaust spring and re-making joint in pipe, 151; commutator, carburettor, and ignition troubles, 108.
10 Horbick. Flaw in combustion chamber, casting repaired by blacksmith, 210.
14 Brooke. Pressure pipe broke inside petrol tank, 158.
20 Winton. Changing battery, cleaning carburettor, and changing plugs and replenishing, 119.
20 Germain. Changing coil, washing valves, and lubricating, 60.
20 Beaufort. Overhauling electrical ignition, adjusting coil and cleaning exhaust valve, and adjusting fan belt, 68; magneto ignition plugs cleaned, fan belt adjusted, tightening joints on water pipe, and tightening nuts holding cylinders to crank chamber, 101.
22 Daimler. Broken tooth in differential gear, 103.

A good many of these are serious matters which could hardly be remedied on the road by an average driver in time to complete an ordinary journey.

In 1906 the corresponding figures are:—

12 Victoria. Carburettor, 62; ignition, 102, 125, 122; carburettor, 67.
18 Arrol-Johnston. Replacing exhaust valve, 141.
20 Maudslay. Repairing clutch, 185.
25 Austin. Key sheared in main sprocket shaft; fitting new one, 70.
20 Horbick. Connecting batteries in place of magneto, 70.

This shows a very considerable diminution over those of 1903, and the most serious delays occurred in two cars which subsequently retired.

Turning now to the question of the number of small stoppages and their causes in 1903, these were as follows:—

Cause.	Number of Stoppages.	Total Marks Lost.	Average Length of Stoppage.
			Minutes.
Ignition, ...	129	1,386	10·7
Replenishing, ...	84	360	4·3
Carburettor, ...	36	358	9·9
Hills, ...	36	137	3·8
Clutch, ...	34	133	3·9
Valves, ...	26	236	9·1
Engines, ...	25	202	8·1
Gear, ...	20	192	9·6
Water circulation, . .	9	125	13·9
Driving, ...	9	18	2·0
Brakes, ...	5	43	8·6
Chains, ...	4	28	7·0
Fans, ...	2	56	28·0
Miscellaneous, . . .	20	128	6·4

In 1906 the figures were:—

Cause.	Number of Stoppages.	Total Marks Lost.	Average Length of Stoppage.
			Minutes.
Hills,	36	209	5·7
Driving,	33	41	1·2
Ignition,	31	228	7·4
Carburettor,	29	156	5·4
Replenishing,	16	37	2·3
Engine,	15	53	3·5
Fans,	8	19	2·4
Clutch,	4	11	2·8
Water circulation,	3	56	18·7
Gears,	2	11	5·5
Valves,	1	10	10·0
Brakes,	1	5	5·0
Chains,	1	1	1·0
Miscellaneous,	13	88	6·7

In these tables the causes of stoppage are inserted in the order of the total number of stoppages caused, as far as can be ascertained by the reports of the trials; but in some cases it is not possible to get this absolutely exactly, as in some stoppages various causes are grouped together and in some cases numerous small stoppages are also grouped. Sometimes the cause of stoppage is not defined. The error due to these causes is not, however, at all material.

In all cases any trouble with petrol supply is put under the head of carburettor.

Where it is quite clear that a stoppage has been caused by the driver either missing his gear or stopping his engine accidentally, this is put under the head of "Driving"; but it seems likely that some small stoppages of this kind are included in the headings of "Engines" and "Gears."

In comparing these tables it is very evident that the number of small stoppages has decreased very largely, especially in the ignition.

Collecting the various stoppages great and small we get *Ignition*:—

	1903.	1906.
Total failures,	1	2
Stops over an hour—Number,	11	4
Marks lost,	1,341	419
Stops under an hour—Number,	129	31
Marks lost,	1,386	209

It is hardly satisfactory that there should be twice as many cars completely fail in 1906 as in 1903, and the fact that both of these were magneto failures and that one magneto, in addition to these, failed while the car continued on accumulators, suggests that these are not yet quite perfect, and that it may be best to fit the better qualities of cars with

two ignitions. On the other hand, there is a great diminution both in the stoppages and in the number of marks lost, while three of the four serious stoppages were in one car, which subsequently retired, and the other was due to the failure of the magneto mentioned above.

Sparking plugs account for a large proportion of the shorter stoppages, a few being caused by commutators. In all the details of the ignition, however, there has been a great improvement since 1903, especially in wiring; there is, consequently, a very satisfactory improvement in the matter of small stoppages.

In carburettor and petrol supply the figures are:—

	1903.	1906.
Total failures,	0	1
Stops over an hour—Number,	1	2
Marks lost,	158	129
Stops under an hour—Number,	36	29
Marks lost,	358	156

This item does not show a satisfactory improvement. The total failure was due to a broken petrol pipe. There are more long stoppages in 1906 and nearly as many short ones, though the marks lost are less in both cases, but considering the much fewer miles run it can hardly be said that there has been any general improvement at all. In a great many cases the trouble was caused by the use of a pressure feed worked from the exhaust, which allowed dirt and water from the exhaust gases to get into the petrol. This suggests that the best way of getting pressure for a pressure feed may be to have a small pump worked by a cam on the engine. This would probably cost no more than the usual pressure feed valve and relief valve. If this is not done there should be some arrangement for preventing dirt getting into the petrol tank. To prevent broken pipes they should not be too small, and should be so arranged that they cannot vibrate.

For engine troubles the figures are:—

	1903.	1906.
Failures,	9	1
Stops over an hour—Number,	1	0
Marks lost,	210	0
Stops under an hour—Number,	25	15
Marks lost,	202	53

This is a very satisfactory improvement. In 1903 there were several failures, broken connecting-rods accounting for some, while in 1906 there is only one from "sand in the engine chamber," which is not very definite, but suggests that the core had not been thoroughly cleared out. There are no

long delays and the short ones are not very important. It may, therefore, be taken that a modern engine kept in good order is quite reliable.

Valves gave the following results :—

	1903.	1906.
Failures, ...	1	0
Stops over an hour—Number, . .	2	1
Marks lost, . .	284	141
Stops under an hour—Number, . .	26	1
Marks lost, .	236	10

This is a very great improvement, and is due largely to the introduction of mechanical inlet valves. It is also due to the fact that several of the competitors in 1903 were using a new brand of petrol which clogged up the inlet valves. Had it not been for this many of the small delays would have been avoided.

Gears and transmission gave trouble as follows :—

	1903.	1906.
Failures, ...	7	4
Stops over an hour—Number, . .	1	1
Marks lost, . .	108	70
Stops under an hour—Number, . .	20	2
Marks lost, .	192	11

This does not at first sight appear very satisfactory, but it must be remembered that the course of the trial of 1906 was much more severe on the gears than that of 1903. The above figures include all failures of transmission gear from the engine to back wheels, except clutches. A careful examination of the details of these failures shows that gears and differentials are still not always up to their work.

Clutches caused one withdrawal in each trial and one delay of over three hours in 1906. On the other hand, there were very few stoppages for slipping clutches in 1906 and a good many in 1903. Water circulation caused two cars to withdraw in 1903, but did not cause any failures or long delays in 1906 and only three short ones; it may, therefore, be considered satisfactory.

With regard to the other items the causes of failure and long delay have been mentioned in detail, and it is not necessary to say more about them. Driving and replenishing stops are not matters of construction, except that the latter are avoided by having good-sized reservoirs for petrol and oil, and enough radiator surface to keep the water from boiling. These matters are now generally satisfactory. Chains and brakes show great improvement, but fans show more stops, though these are very short. Miscellaneous stops are much fewer, due to better construction of small details and better locking

of nuts. The latter is a most important point, many stoppages in 1903 being due to the nuts not being properly locked.

An examination of the particulars of the cars entered for the trials will also tell us the tendency of the changes of construction during the last three years. In this respect there is no very startling change in the general outline of design, which follows very closely that of 1903.

The most important point which strikes one in examining the particulars of the cars in the two trials is that there has been a very great increase in the average weight of the cars. Taking the cars in Table iii. it will be seen that the heaviest car is the 22 Daimler, No. 136, which weighed 32·6 cwts. This, however, was a car to carry ten people, and the heaviest four-seated car weighed 26·3 cwts. On the other hand, the lightest four-seated car weighed 11·4 cwts. Two-seated cars weighed anything down to under 8 cwts. The car which made the best average speed only weighed 17·9 cwts., and it had 251 cubic inches cylinder capacity. The majority of the larger cars weighed something in the neighbourhood of a ton without their passengers, while the smaller four-seaters were about 15 cwts.; and there were two four-cylinder four-seated cars under 16 cwts., though these are not in the table for the reasons given. When we turn to the Scottish trials of 1906 we find a great increase. One four-seated car weighs 34·5 cwts., or about 2 cwts. more than the ten-seated car of 1903, and 8 cwts. more than the heaviest four-seated. Several weigh over 30 cwts. In the smaller cars the same appears. The lightest car is 11·5 cwts., and is an American one; and the lightest European car which got through the trials is 12·5 cwts., and carries two people only. Several of the two-seated cars weigh over 14 cwts. The lightest four-seated car is over 16 cwts., and the lightest four-cylinder car 19·2 cwts. The general average of the four-seated cars is well up to 25 cwts. instead of being under a ton.

This is a matter which requires explanation. In some cases it is simply a case of general increase of the size of the car and engine. The largest cylinder capacity in 1903 was 366 cubic inches, while in 1906 it had risen to 525, and in several cars it was over 400, and in many it was over 300. This simply means that there is a demand for more powerful cars than were wanted in 1903, and the lightness can only be compared by comparing the cylinder capacity per cwt. The highest cylinder capacity per cwt. loaded in 1903 was 11·1, while in 1906 it was 13·2. On the other hand, in the medium-powered four-cylinder cars there seems to be an increase in weight without any increase in power, as the majority of the four-cylinder cars in the 1903 trial had 9 cubic inches of cylinder capacity per cwt., and many of the 1906 cars fall below this. The cause of this is, no doubt, largely the demand for more roomy and comfortable bodies and the lengthening of the wheel base in order to have a side entrance. In addition to this it may be admitted that, in some cases, parts of the cars of three years ago were found to be too light for the work they had to do. Making all allowances for these points, however, it seems that the weight of many of the modern cars is excessive, and an examination of the particulars of the cars favours this view. For instance, the Rolls-Royce car, with 242 cubic inches of cylinder capacity and 8 feet 10 inches wheel base, weighs 19·25 cwts.; while other cars with a considerably smaller cylinder capacity and about the same or shorter wheel base weigh considerably more. Further, in the case of the two-seated cars, the accommodation is very much what it was three years ago, yet the weight is often pretty

well double. Many of the comparatively light cars of three years ago are still running satisfactorily, so it can hardly be accounted for by necessary increase in strength. Knowledge of motor construction should also have advanced enough in three years to enable us to remedy such weaknesses as they had without increasing the weight materially. An examination of a good many cars gives one the impression that weight is not considered nearly as much in their construction as it should be, especially in some of the details and methods of construction. The general use of ball-bearings has possibly also added to the weight, especially in connection with the carrying of the weight in live axles on the tube, instead of on the revolving part.

At all events it is a matter which will require serious consideration, both on the part of those who buy cars and those who make them, as the running cost of a car depends very largely on its weight. In the larger cars this will not be a serious matter, as those who buy them will, in most cases, be rich men who can afford the expense; but those who buy moderate-priced cars will generally be people to whom cost of running is of great importance, and as the expense of tyres and petrol increases rapidly with the weight of the car this must be kept down. It might, in fact, be better to renew wearing parts a little more frequently, rather than have excessive tyre repairs.

In comparing the petrol consumption in 1903 and 1906 we see the effect of the increase in weight. This is shown both in the actual car miles per gallon and also in the passenger miles per gallon. The latter is the real test of the car's performance. A comparison of the latter will show that the performance of the 1906 cars is not nearly as good as those of 1903. In the latter 80 passenger miles per gallon is quite common; in the former many of the cars fall below 60.

The ton-miles per gallon in 1906 shows an improvement, as one would naturally expect, from the greater weight of the cars, but it is very small. The average consumption of all the cars which finished shows an improvement of about 5 per cent. only over 1903. It is difficult to make any good comparison between the two trials on this basis, as the routes were different; but when we consider that the marks given for petrol consumption in 1903 were very few, and therefore in many cases little care was taken to economise it, it is doubtful whether there has been any improvement.

Much may be learnt from a careful study of the trials as to the relative advantages of different methods of construction, but to do this we require many details of construction which are not given in the report, and also a space for consideration, which cannot be granted here. A few points may be noticed.

With regard to the relative reliability of low-tension and high-tension ignition we are in a difficult position, as in some cases it is not quite clear what ignition cars were fitted with, and, besides this, some cars were fitted with two ignitions. In this case, if one failed, the car might continue on the other without any failure being reported. In regard to the absolute breakdowns, one car with high tension and one with low tension were withdrawn, the cause of failure being in each case given as the magneto failing. In addition one high-tension magneto failed completely, and in another case where a car was fitted with two high-tension ignitions, one apparently failed and the journey was continued on the other. As far as can be ascertained, there were about twenty cars fitted with low tension and sixty with high tension. As there was one complete failure with

each system, the percentage of failures would be higher with the low than with the high, but one failure is too small a number to draw averages from. To get the proportion of failures right we ought also to know of all the failures of each system when two are installed on one car, but of this there is no evidence. In the matter of delays, however, there seems to be a great advantage in the low tension. The high-tension cars had a total of 31 delays, totalling 619 minutes. The low tension had 3 delays, totalling 9 minutes. The average delay per car with high tension was 10·3 minutes and with the low tension ·5 minute.

With regard to the percentage of cars which ran through with no ignition trouble we have to eliminate cars which broke down from causes outside ignition. Excluding these, 70 per cent. of the cars with high tension had no trouble and 83 per cent. of those with low.

The obvious lesson from a study of these figures is that ignition is now getting very reliable, and that a really well installed system, whether high or low, can be made to give very good results. The figures show a very great advantage to the low tension, but there are not enough data to generalise on, and what there is may not be quite accurate, as mentioned. Still, allowing for this, there seems to be an advantage to the low tension. We may allow that it was used in the better class cars and the high tension in many of the smaller and cheaper ones, and that, therefore, the details might be better carried out in the low tension; but there are a good many failures in the high tension which can hardly be accounted for by this. One lesson is that, when two high-tension systems are provided, they should each be completely separate, with their own sparking plugs, as there were many failures of these.

With regard to the low tension there was only one failure of any importance, and this shows that a low-tension system should be made thoroughly reliable. The failure is given as "contact on magneto breaking," and I have assumed that it was the magneto itself which failed, but it may mean the contact in the cylinder. In either case, it should be avoidable, as the magneto itself should *not* fail; while contacts in the cylinder should be as easily replaceable as a sparking plug.

Two points on which we ought to be able to get information are—(1) the value of four speeds as against three; (2) the value of ball bearings.

With regard to the first, there does not seem the least evidence of any advantage at all in four speeds over three. Theoretically, of course, the four speeds should enable us to get more power out of a given engine and ought to show this in the hill-climbs. To sift this matter thoroughly a complete comparison of all the cars would have to be made on the basis of their weights and cylinder capacity, but this is not possible here. Beginning in 1899, however, it is demonstrable that the three-speed cars have had enough successes to show that if there is any advantage in the extra speed it can only be very small. A glance will show that in 1899 the cars with three speeds or less were on the average far faster than those with four. Coming to later times, in 1903 the best results in terms of horse-power per 100 cubic inches cylinder capacity were made by three-speed cars, though the actual fastest times were made by four-speed cars, as there were no three-speed cars of anything like equal power to the four-speed.

In 1905 the fastest average speed was made by three-speed cars over two hills of such varying gradient that changes of speed should have had every opportunity of proving their advantage, and the best horse-power per 100

cubic inches of cylinder was also shown by a three-speed car. In 1906 the second fastest car, taking the mean of four hill-climbs, had three speeds, and in the only two classes in which there was a fair proportion of three- and four-speed cars the best times seem to have been done by three-speed ones.

With regard to ball-bearings we get little information. In 1905 both the best actual time and best horse-power per 100 cubic inches of cylinder capacity were done by plain bearing cars, and this shows pretty conclusively that if there is any advantage in the ball-bearing it cannot be very material. By 1906 the ball-bearing had become too general for comparisons to be made, but if there was any great saving in friction it would have shown itself in the petrol consumption, and this has already been dealt with.

Petrol Consumption. Ton-Miles per Gallon.	Petrol Consumption. Car-Miles per Gallon.	Petrol Consumption. Passenger-Miles per Gallon.	Diameter of Back Wheels.	Ratio of Gear. 1.	Ratio of Gear. 2.	Ratio of Gear. 3.	Ratio of Gear. 4.	Tractive Force. Lbs. per Ton. 1.	Tractive Force. Lbs. per Ton. 2.	Tractive Force. Lbs. per Ton. 3.	Tractive Force. Lbs. per Ton. 4.	Passengers Carried.
			Inches.									
11·7	67·0	67·0	26	...	...	...	12·7	...	...	...	490	1
23·3	22·2	44·4	...	Four speeds.				...	...	...	...	2
39·5	33·4	133·6	43	Three speeds.				...	...	...	...	4
10·0	15·4	30·8	30	Three speeds.				...	...	...	...	2
12·6	15·7	31·4	38	...	...	...	...	...	...	...	...	2
42·0	40·0	160·0	35	14·4	...	...	5·5	420	...	...	162	4
9·9	16·7	33·4	...	Three speeds.				...	...	...	...	2
18·3	11·8	70·8	39	Three speeds.				...	...	...	...	6
23·3	22·2	88·8	41	Three speeds.				...	...	...	...	4
21·4	28·6	57·2	37½	Two speeds.				...	...	...	...	2
20·4	40·0	80·0	36	Three speeds.				...	...	...	...	2
17·5	33·3	66·6	30	11	...	...	5	490	...	...	223	2
41·0	33·3	133·2	42	20	10	6½	5	356	178	115	89	4
10·7	14·2	28·4	38½	Three speeds.				...	...	...	...	2
10·8	30·8	61·6	25	Three speeds.				...	...	...	...	2
...	20·0	100·0	...	Four speeds.				...	...	...	...	5
33·3	22·2	133·2	39	24	12	8	6	375	188	124	94	6
24·0	20·0	80	41	23	13	7·7	5·6	378	215	128	92	4

												Load Carried.
												Cwts.
47·0	5·0	...	47	60	38	21·5	12·5	300	188	107	62	120
34·2	7·3	...	...	...	...	...	...	...	...	...	...	44
27 0	6·7	...	42	32	16	11	9	171	85	59	48	36

No.	ty N ed.	Passengers Carried.	Speed in Miles per Hour.		Petrol Consumption.			Mean Horse-power on Hills.	Horse-power per 100 Cubic Inches Cylinder Capacity.
			River Hill.	Westerham Hill.	Ton-Miles per Gallon.	Car-Miles per Gallon.	Passenger-Miles per Gallon.		
1	Hun	1	21·1	...	11·7	90·0	90·0	1·32	6·3
88	Pan	4	16·5	11·3	28·7	23·4	92·6	9·4	4·5
69	Wol	4	13·7	12·5	17·7	10·4	41·6	12·4	3·9
83	Pas	4	15·3	10·5	15·3	9·2	36·8	11·7	3·3
86	Dai	4	13·0	11·6	24·0	14·5	58·0	11·4	4·2
75	Clém	4	12·4	11·3	16·0	14·5	58·0	7·3	6·2
74	Ger	4	12·3	10·2	19·8	14·3	57·2	8·8	4·0
64	Peu	4	13·9	7·8	23·6	18·2	72·8	7·6	4·7
51	Glad	4	10·8	8·4	18·7	17·7	70·8	5·5	4·8
24	De I	3	9·9	7·6	18·7	30·0	90·0	3·3	7·9
76	Dai	5	9·7	7·7	21·0	14·4	72·0	7·2	5·1
84	Pas	4	8·7	8·3	21·2	13·7	54·8	7·5	2·1
82	Mau	4	9·5	7·3	19·7	12·5	50·0	7·3	3·0
5	Peu	2	9·3	7·3	18·3	36·3	72·6	2·3	4·8
44	New	4	8·8	7·2	18·3	20·2	80·8	3·9	3·7
41	Wol	4	8·2	7·6	22·3	17·4	69·6	5·7	3·6
T1	Pan	...	8·6	7·0	...	...	...	...	...
52	Arie	4	8·7	6·5	20·0	18·7	74·8	4·1	6·2
T12	Nap	...	12·2	3·0	...	...	...	...	...
54	Cent	4	8·7	6·2	28·2	24·8	99·2	5·0	6·0
47	De I	4	8·3	6·5	20·5	20·8	83·2	3·7	6·5
66	Hun	4	7·8	6·9	22·4	16·5	66·0	5·6	3·6
T4	Wol	...	7·8	6·4	...	...	...	...	...
30	Deca	2	7·3	6·7	14·0	13·6	27·2	4·2	5·5
33	Glad	4	9·2	4·6	20·2	20·0	80·0	3·3	3·9
T7	Nap	..	8·3	5·5	...	...	...	...	...
32	Jam	4	7·3	6·4	21·5	17·9	71·6	4·6	3·1
81	M.M	4	10·7	2·5	14·3	10·6	42·4	4·6	1·8
23	M.M	4	6·7	5·8	19·4	23·6	94·4	2·8	4·5
42	Bels	4	6·8	5·7	22·0	17·6	72·4	4·1	3·7
57	M.M	4	6·0	5·8	18·0	15·8	63·2	...	...
T8	M.M	...	6·4	5·1	...	...	...	...	...
71	Wils	4	8·3	2·3	18·9	14·2	56·8	3·9	2·7
35	Broo	4	5·5	5·3	13·6	13·5	54·0	4·0	2·7
59	Germ	4	6·1	4·5	17·6	14·3	57·2	3·6	3·2
65	Brus	4	8·5	1·3	16·0	12·8	51·2	3·0	1·7
19	Star,	4	5·3	4·3	19·3	18·2	72·8	2·8	3·3

Notes on Table I.

Transmission.—The Panhard, Daimlers, and M.M.C. had gear and chain drives generally like those of a modern car, but with shaft-to-shaft drive.

Barrière tricycle; driven with gear with a countershaft between the engine and road wheels.

Critchley, Daimler, and Leigeoise; belt and gear.

Benz, Hurtu, Delahaye, Valee, Mors, Ducroiset, and Bergman; with belt and chains.

Lanchester; worm and wheel, with direct drive on top speed.

Ignition.—Barrière tricycle; single-contact high-tension.

Benz, Hurtu, Delahaye, Valee, Ducroiset, and Leigeoise; high-tension, with ordinary trembling coil, low-speed trembler.

Mors; low-tension, with accumulator to start and dynamo to run.

Lanchester and Bergman; low-tension magneto.

All Daimlers and Panhard; tube.

Tyres.—Barrière, Daimler phaeton, Critchley, Lanchester, Hurtu, Delahaye, Panhard, Valee, Mors, and Leigeoise; pneumatics.

M.M.C., Daimler waggonette, Benz, Ducroiset, Bergman, and Canstadt-Daimler; solid rubber.

The German lorries had iron tyres and were driven entirely by gear, but with clutch and change-speed gear much as other Daimlers. The Post Office van had solid rubber tyres and chain drive.

The weights are those stated by makers, and may not be quite accurate. The general tendency of makers seems to be to understate their weights, and should this be the case here the horse-powers will not be accurate.

The tractive forces are taken from the speeds given by the makers, but these also may not be accurate, and they will be affected by the weights. 80 lbs. mean pressure has been assumed and 10 per cent. loss on each gear drive as far as the number of these is known. Where the tractive force is not given it is because the ratios of gears or weight are not available.

Where the cylinder dimensions are given in millimetres and inches it is because the former are the dimensions in the official report, and in this case the latter are only approximate.

In this and all other tables where there are less than four speeds, the top is entered under the heading of the "fourth," so that all the top speeds shall be in one column and all the bottom in one column, the intermediate speeds being omitted. This seems much more convenient for comparison than omitting the fourth when there are only three and putting the top in the "third" column, as it enables comparisons of the top and bottom speeds to be readily made, while the intermediate speeds are of less importance.

Notes on Table II.

In this table the cars are arranged in order of the mean speed they made up River Hill and Westerham Hill. Only cars which completed the trial and took all their load up the hills are included. The Humber bicycle was pedalled on Westerham Hill.

The 2 horse-power Humber, 5 Century, 4½ Renault, 7½ Wolsey, 22 Daimler, and 8 Clément finished the trial satisfactorily, but failed to take their load up one or other of the hills without assistance.

Nine other cars started, but were withdrawn or broke down.

The cars with T before their number were entered in the amateur class, and, therefore, there are no particulars of their performance.

The petrol consumption in this trial was taken on a comparatively short distance forming part of one day's run, and possibly there is some local cause for all the petrol consumptions being rather heavy.

Notes on Table III.

In this table the cars are arranged in the order of their mean speeds, taking the mean of the four hill-climbs given, and also the run on the flat. Only cars which took all their load up all the hills without assistance and also completed the trial are given. The particulars are much the same as in previous tables, but some extra columns are added giving details based on the ratios of speeds. This is not taken from the official report and may not be exact, but it is taken from data published at the time on the makers' authority and the errors in it are probably not very great. Four of these columns give the speed of the car at 1,000 revolutions per minute of the engine. All the speeds are reduced to this, so that the relative speed of the cars at a given number of revolutions can readily be made, but it is not suggested that all the engines will habitually be run at this speed, or that they will all run at the same speed. It is, however, a more convenient way of putting it than to give it at the "normal revolutions" as stated by the makers, as in practice these have little to do with the speeds the engine is run at on the road.

This will be seen from the columns which give the revolutions of the engine (1) going up Westerham Hill, and (2) on the flat at Bexhill.

It is assumed that the whole of the Westerham run was done on the bottom and the whole of the flying kilometre on the top speed. It is possible that this is not correct, and that some of the cars went up part of the Westerham Hill on their second and that some of them ran the kilometre without using their top speed; in this case the revolutions given for Westerham Hill would be too high and for Bexhill too low.

There is enough evidence here, however, to show that the majority of competitors ran their engines at their utmost speed, and that the engines (such as the De Dion, &c.) which would run the fastest did best in proportion to their cylinder capacity.

The tractive forces are worked out as before on the assumption of 80 lbs. mean pressure, but as the exact nature of the drive is not stated they are only very approximate. The calculations, however, agree fairly well with the observed facts, as the steepest part of Westerham Hill is a gradient of one in 7·2, and so the tractive force required should be about 350 lbs. per ton. It will be seen that all the cars which got up had this or more, and one or two which had slightly less had to go up on their reverse, which was probably a lower gear than their low speed forward.

The following cars completed the trial satisfactorily, but failed to take all their passengers up Westerham Hill without assistance; 24 and 12 Richard, 10 and 20 Thornycroft, 16 Argyll, 13 Rex, 12 Relyante, 10 Deschamps, 6 Regal, and Century Tandem. Many of these cars had ample

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No.	'apacity /t. Loaded.	Best Horse-power at Road Wheels.	Horse-power per 100 Cubic Inches Cylinder Capacity.	Passengers.	Wheel Base. Ft.	Wheel Base. Ins.	Track. Ft.	Track. Ins.	Miles per Gallon of Petrol.	Passenger-Miles per Gallon.	Ton-Miles per Gallon.
10	5·7	5·6	7·0	2	5	5½	4	1	28·3	56·6	19·4
22	3·3	4·9	11·7	2	6	0	3	9½	30·3	60·6	19·1
18	5·7	5·1	6·4	2	5	5½	4	1	38·5	77·0	27·5
14	3·5	4·9	10·2	2	5	10	3	9	38·8	77·6	26·2
15	5·7	5·6	6·5	2	5	10	3	8½	29·5	59·0	22·3
26	3·7	5·2	9·8	2	5	9½	3	8½	23·6	57·2	17·0
20	5·8	4·8	6·0	2	5	5½	4	1	24·6	59·2	17·1
30	...	5·7	...	2	7	4	3	11	33·3	66·6	27·5
38	5·0	5·4	6·4	3	5	11	3	8	25·4	76·2	21·6
23	3·3	4·3	10·2	2	6	0	3	9½	30·3	66·6	19·1
32	3·7	5·1	9·0	3	5	6	3	8½	34·5	103·5	26·2
24	...	5·8	...	4	5	11	3	8½	18·4	72·8	17·1
34	5·1	4·6	6·0	2	6	2½	3	11	16·9	33·8	13·0
17	4·2	3·8	6·7	2	5	6	3	8½	33·3	66·6	22·9
25	6·7	6·2	4·3	3	6	8½	4	7	15·6	46·8	16·6
16	4·6	3·8	6·7	2	5	3½	3	6	37·9	75·8	23·6
13	4·3	3·7	6·2	2	5	11	3	9	24·1	48·2	16·6
37	3·6	4·5	7·5	2	6	7½	4	2	27·2	54·4	22·8
31	...	...	...	3	6	0½	3	8	20·1	60·3	16·7
8	2·9	3·4	8·1	2	6	1	4	0	23·3	46·6	17·0
11	...	3·0	...	2	5	10½	3	8½	25·3	50·6	17·8
12	3·9	4·0	6·6	2	6	0½	4	0	31·4	62·8	26·6
1	3·5	3·2	7·6	2	5	3	3	9	28·5	57·0	17·0
2	3·6	3·2	7·6	2	5	2½	3	8	25·8	51·6	15·1
4	3·3	2·9	6·9	2	5	3	3	9	24·4	48·8	15·5
21	...	3·5	...	2	5	10	4	0	47·2	94·4	29·7

	ts.	Cylinder Capacity per Cwt.		Passengers Carried.	Petrol Consumption.			Wheel B
	l Loaded.	Light.	Loaded.		Ton-Miles per Gallon.	Car-Miles per Gallon.	Passenger-Miles per Gallon.	
		Cubic Inches.	Cubic Inches.					Ft. In
Daimler,	34·3	15·9	12·8	4	21·4	12·5	50·0	8 8
Darracq,	33·2	15·9	12·6	4	28·5	17·1	68·4	9 1
Beeston	34·9	14·0	11·6	4	20·0	11·5	46·0	9 11
Benz, .	35·7	12·2	10·1	4	22·8	12·8	51·2	10 3
Ariel, .	41·0	12·5	10·4	4	22·2	10·8	43·2	9 7
Richard	35·6	10·9	8·7	4	31·2	17·7	70·8	9 6
Armstro	35·8	10·7	8·8	4	27·0	15·1	60·4	9 9½
Ariel, .	38·8	12·0	10·0	4	21·3	10·9	43·6	9 7
Ariel, .	39·7	15·8	13·2	4	24·0	12·1	48·4	9 10
Mass, .	31·6	10·9	8·9	4	25·5	16·2	64·8	10 0
Rolls-Ro	25·6	12·5	9·4	4	27·4	21·4	85·6	8 10
Rolls-Ro	33·1	13·6	11·0	4	23·7	14·3	57·2	9 8½
Mettalur	34·2	11·8	9·1	5	28·2	16·5	82·5	9 6
Straker	35·0	10·6	8·6	4	26·9	15·3	61·2	9 6
Iris, .	31·8	12·0	9·7	4	20·5	12·9	51·6	9 6
Belsize,	35·9	11·5	9·3	4	28·6	15·9	63·6	10 1
Pipe, .	31·4	9·1	7·3	4	24·4	15·5	62·0	9 9
Germain,	26·8	8·6	6·6	4	26·6	19·8	79·2	8 8
Martini,	38·5	13·4	11·2	4	[illegible]	[illegible]	[illegible]	[illegible]
Horbick,	2·9	11·3	9·0	4	...	...	...	10 0
Peugeot,	9·3	11·2	8·8	4	...	...	...	8 7
	8·3	11·1	9·4	4	...	...	...	9 0
Ridley,	..	...	...	2	...	...	...	5 4
Leader,	..	...	...	4	...	...	...	7 6
Marchand,	..	...	...	4	...	...	...	8 10
Leader,	..	...	...	4	...	...	...	10 1
Calthorpe,	..	...	...	5	...	...	...	8 7

ed hill-climbs.

power, so it seems probable that they did not have a low enough bottom speed.

The horse-power in this table is taken from the Club Report and apparently included a considerable allowance for friction in the transmission gear.

Notes on Table IV.

The particulars in this table are much the same as before, except that several cars are included which failed to get up the trial portion of Fromes Hill. A comparison of their speeds up the other hills with those of the cars which actually got up very clearly shows how misleading the results of a hill-climb may be, as up Dinsmore Hill several of the cars which stuck on Fromes Hill are faster than some of those which got up. Had the hill-climbs on Dinsmore Hill been taken alone they would have given a purely fictitious value to the performance of cars which were too highly geared on their low speed to get up Fromes Hill. Even the latter is given as being only about 1 in 8 on the steepest part used in the hill-climb, so that on a steeper hill some of the cars which did very fast times on Fromes Hill might easily stick. In fact, it appears that several of these had to shed passengers at various parts of the trial, though not on the trial hills.

The horse-power at road wheels is taken from the Club Report, and appears to include an allowance for friction of transmission gear. The horse-power given is that of the best performance of each car on any one of the three hills.

It will be noticed that the only car in the table of standard track is the Oldsmobile, whose track is 4 feet 7 inches. This car is built in America.

Notes on Table V.

General particulars as before. Cars are arranged in the order of their mean speed up two hills, viz. :—Cairnwell Hill, 1,226 yards long, with a mean gradient of 1 in 8·9, and Loch-na-Craig Hill, 3 miles 484 yards long, with a mean gradient of 1 in 17.

In taking horse-power at the road wheels no allowance is made either for wind resistance or friction of the transmission, but allowance is made for road resistance.

The following cars completed the trial satisfactorily, but failed to take their full load up all the hills :—8 Darracq, 12 Sunbeam, 24 Ryknield, 12 De Dion, and 7 Vauxhall. The last two failed on one of the above-mentioned trial hills and the remainder on other hills in the course of the trial.

Notes on Table VI.

In this table, as it deals with modern cars, particulars of all the cars entered are given, but those relating to the cars which failed wholly or partially are given separately.

The successful cars are arranged as far as possible in the order of their mean speeds on the four hill-climbs, but no actual times or speeds are published for these, and as the cars are marked separately for the different classes there may be slight errors in this. As no times or speeds are published the hill-climbs lose a great deal of their technical value, as without

these no conclusions can be drawn and there is a difficulty in comparing the relative merits of the cars.

Four cars which succeeded in taking their load up the trial hills failed to take it up other hills on the route, as noted. In this table three weights are given. First, the approximate weight as stated by the makers; second, the actual weight of the car without passengers; and third, the weight with passengers.

Notes on Table VII.

This table gives particulars of some of the leading cars of 1906 as stated by the manufacturers. In the majority of cases the dimensions speak for themselves and no comment is necessary.

It may, however, be pointed out that in many of them the wheel base can be varied within considerable limits to suit different bodies, and, therefore, that given would only be correct with one particular type of body.

In the matter of wheel track it will be noticed that some makers adopt the plan of having one standard wheel track for all their cars, whatever size they are; while others have a great variety of tracks for different-sized cars. It will also be noticed that in the majority of cases the tracks of the better class cars are the ordinary standard—*i.e.*, between 4 feet 6 inches and 4 feet 9 inches—while in many of the smaller and cheaper cars they are very much narrower than this; and that the only two cars of American origin, the Cadillac and Duryea, have the standard track, as in all American cars. The advantage of this in saving tyres has already been pointed out.

The weights given are those stated by the makers, but naturally will depend entirely on the body fitted. In many cases customers now require very heavy bodies, which would make the weight of the cars considerably more than is stated in the tables, particularly in the case of the larger cars. See the weights of the cars as actually taken in the Scottish trials.

It will be seen that the highest class makers generally give all the cars they build wheels of reasonable size. Panhard and Dietrich, for instance, use nothing smaller than 34 inches for their smallest cars, even of 8 horse-power. There is no doubt that many of the smaller cars have much too small wheels, and that even the larger ones would be much better with larger wheels. Many of the older cars had 40-inch back wheels, but these were abandoned because it was desired to make the front and back tyres interchangeable, and it is not convenient to put in very large front wheels on account of the difficulty of getting enough lock. It is now common, however, to have the front and back tyres in large cars of different size, and in this case there is no advantage in having the wheels the same, and 40-inch wheels would be a great advantage. For smaller cars it is much more convenient to have the wheels the same size so as to have the tyres interchangeable.

Notes on Table VIII.

The particulars in Table vii. give data for calculating other elements of the various cars, some of which are given in Table viii.

The first column indicates the ratio of stroke to diameter of cylinder. This, as will be seen, varies from ·75 in the Lanchester to 1·6 in the smallest Renault. There is a general tendency to make the stroke relatively shorter

6	...	...	4½	5
8	...	...	4	4
12	...	...	4½	5

in the larger engines, but there seems to be no good reason for this, as good results are obtained with all the different proportions.

The next column shows the total cylinder capacity in cubic inches, this being the real measure of the bulk of the engine.

The next four columns give the speeds of the cars at 1,000 revolutions, in order that the speed they are geared to can readily be compared.

The next gives the weight of the car loaded with its full complement of passengers, allowing 1½ cwt. for each passenger; and the next two columns indicate the cylinder capacity per cwt., both light and loaded, this being generally really the best comparison for power and weight. This would, of course, be modified by the type of body used and its weight. From the speeds and revolutions the ratio of the gear between the engine and back wheels can be calculated; this is given in the next four columns. It will be seen that in the majority of cases gear-ratios of over 4 to 1 are not used on the top speed except for the very small cars which have to run their engines very fast to get the necessary power. A comparison of the gear-ratios, cylinder capacity per cwt., and sizes of wheels shows what an advantage it is to have a good proportion of cylinder capacity. Otherwise a very large ratio of gear is needed for the necessary tractive force, or very small wheels. The disadvantages of high gear-ratios and small wheels have already been referred to, and when it is considered how little it costs to make an engine larger, it seems that it would be better to do so in many cases.

The next column gives the ratio between the top and bottom speeds and the next four, the tractive forces calculated on the basis of 80 lbs. mean pressure and 10 per cent. loss in each gear. This is, of course, only an approximate comparison, as the actual efficiency of engines and gear vary. Further, it is very evident on examining the tractive forces that in some cases the cars would have the gear different from that stated for particular purposes. In fact, it must be remembered that the ratio of gear from the engine to back axle can be easily altered, and would be so, to suit the ideas of the customer.

A careful comparison of the figures shows several things, however. It will be seen that in most of the cars there is a well thought out scheme of tractive forces, and that many of them provide over 500 lbs. per ton as their normal amount on their low gear. Some go a good deal above this, but there hardly seems any great advantage in having a tractive force of much over 700 lbs. per ton. Probably in most cases where the table shows tractive forces of over 700 lbs. per ton or under 400, the speeds given would be altered in actual practice according to the weight of the body supplied. We can, however, see from the tables that for small cars with moderate powers a ratio of at least 3½ to 1 is necessary between the top and bottom speeds in order to ensure the necessary speed on the top and hill-climbing power on the low. Many of the cars provide a ratio of over 4 to 1; this may in many cases be better still. On the other hand, it seems as if the ratio in the higher-powered cars might be reduced with advantage. In these the cylinder capacity per cwt. is enormously more than in the low powers, and, therefore, to use the extra power we must either go faster on our top speed or else climb steeper hills without changing speed. In either case a very much higher tractive force is required than for the lower powered car, as greater speed means an increased wind resistance to be provided for, while steeper hills can only be ascended by increasing the tractive force. A higher

tractive force on the low speed than that given by the lower powered car is not needed, as all modern cars should be able to go up any hills on which the wheels will grip. Thus it is, probably, not worth while having a very powerful car, unless its tractive force on its top speed is well over 250 lbs. per ton and 300 might be better; otherwise the engine may never be able to develop its full revolutions on its top speed. Then with a 4 to 1 ratio we have from 1,000 to 1,200 lbs. per ton on the low speed, which is unnecessary as it means that all hills, however steep, can be climbed on the second. On the other hand, there is no necessity to have numerous speeds with very small differences between them; hence for a powerful car four speeds are quite unnecessary as is manifest from the fact that the large majority of races have been won and records made by cars with three and even with less than three speeds. On the other hand, for small-powered cars four speeds might be needed, but here considerations of cost come in, and the money spent in a fourth speed would be much better spent in a larger engine.

This principle is carried further in the Duryea car, which has only two speeds, but has 16 cubic inches of cylinder capacity per cwt. when loaded. This car is speeded to 40 miles an hour at 1,000 revolutions, which is fast enough for all practical purposes, and should be able to run much more than this, temporarily, if necessary. It has 300 lbs. per ton tractive force on its top speed, and, therefore, will go up anything on this that most cars will on their second. It has a low speed for still steeper hills. What are more speeds wanted for?

The last column gives the number of cubic inches of cylinder capacity allowed per nominal horse-power. This is not very instructive technically, as the rating of engines is more a matter of commercial consideration than an expression of the power they will actually do.

Notes on Table IX.

This table gives particulars of some engines considered to be of interest, which, for various reasons, are not included in Table vii. It includes some engines, such as the smaller De Dions, which are practically obsolete, but did good work in their day; some, such as Aster, which are used in cars, but are made for the trade only; the Antoinette engines, which have been so successful in racing boats; and some of the larger size marine motors, as the Thornycroft, Wolsey, &c., and the Beardmore marine gas engine. Also some air-cooled bicycle engines.

The list might be enormously extended if space permitted, but to give particulars of all the successful engines would be impracticable here.

Notes on Table X.

The history of the racing car for the last ten years has been simply a history of the development of engine power at the expense of other qualities. The early racing cars were little more than touring cars, and were fitted with touring bodies, &c., and all the parts were made generally of the same proportions and margin of strength as the touring cars, of the time. Gradually, however, the engine-power developed, and the racing cars became machines built purely to develop the highest possible speed for the duration

N.	Weight.			Cylinder Capacity per Cwt.	Mean Speed.	
	Chassis.	Complete Car.				
		Light.	Loaded.			
	Cwts	Cwts.	Cwts.		Miles per hour.	
Rolls	13·6	18·0	24·7	9·9	39·6	
Berli	17·2	21·8	26·5	8·2	35·4	
Darra	14·4	19·8	26·4	7·1	34·1	
Clém	17·7	22·2	29·8	7·6	33·6	
Beest	13·4	17·8	24·6	8·5	32·6	
Cove	17·0	22·1	28·2	7·1	32·1	
Arrol	14·9	20·3	26·4	8·7	30·0	
Sidde	12·2	17·2	23·6	8·5	27·8	
Scout	15·3	19·4	27·4	6·5	23·1	
Argy	13·1	17·0	24·4	7·9	...	Finished, but only carried part of load round last two rounds.
Berli	15·4	19·4	26·6	8·7	...	160·1 Run out of petrol after running this number of miles.
Thor	16·1	21·0	28·1	5·7	...	148·3
Vino	17·3	21·0	28·7	7·8	...	146·8
Darr	14·7	20·0	26·0	7·2	...	145·8
Hard	19·5	24·0	30·4	10·4	...	138·8
Acad	15·1	18·1	26·1	4·2	...	131·8
S.C.	19·5	23·8	30·4	5·1	...	108·1
Star,	17·2	22·0	28·8	8·7	...	104·6
Bian	17·3	22·0	29·5	9·4	...	101·6
Star,	17·1	21·9	28·7	8·7	...	98·6
Jame	17·2	20·6	24·4	10·2	...	Cracked cylinder jacket.
Mine	18·8	23·4	31·0	7·3	...	Broke clutch on 3rd lap.
Rolls	13·8	18·1	24·8	9·8	...	Broke front spring on 1st lap.
Mine	19·4	22·7	30·8	7·3	...	Burnt clutch leathers on 1st lap.
Sidd	12·0	17·0	23·8	8·4	...	Broke radius-rod on 2nd lap.
Clim	16·1	20·1	27·3	4·0	...	Broke crank shaft on 2nd lap.
Swif	15·8	19·8	27·0	7·2	...	Broke gear on 2nd lap.
Deas	19·1	23·8	31·0	8·9	...	Broke gear on 1st lap.
Vici	16·4	20·5	27·8	6·0	...	Broke steering gear on 1st lap.

of the race they were built for, regardless of any considerations of durability or comfort.

This is carried out furthest in the machines built for the short distance records, such as the 200 Darracq, which holds the record for speed over a short distance, its speed being over 120 miles an hour for the 2 miles. In fact, to all interested in the racing cars the Darracqs are very instructive cars to study. In them all non-essential parts are done away with, and the car consists of little more than an engine and two axles, with a very light frame to carry them. There is no bonnet, dashboard, floor boards, or gear box. There is no differential, so the change-speed gear can be put on the back axle, and there are, in some cases, only two speeds. There is no real body, but only two seats which, in some cases, form the water tank.

Particulars of some of the racing cars are given in Table x. as far as they can be ascertained, but in some cases makers have built several racing cars differing slightly from each other in the same year, and it is not quite clear that the particulars published are those of the cars actually run in the races, as there is naturally a good deal of reticence about the matter.

The cars for 1906 are those which were entered for the Grand Prix. With one exception all these cars had the cylinders of cast iron cast in pairs; the Bayard and Panhard had separate steel cylinders. All the cars had magneto ignition either high- or low-tension. All had centrifugal pumps, except the Gregoire and Renault, which had a thermosyphon.

It will be seen that, while a few years ago four speeds were universal, there is a general tendency to come down to three, makers having concluded that it is better to put the weight into the engine than the gear box. The majority of the cars had wooden wheels, though a few, including some of the most successful, had wire wheels.

It is not possible here to go into the results of the season's racing. Briefly, for speed the 200-horse eight-cylinder Darracq holds the record, having covered two miles in less than a minute.

Of the three International long distance races, the Grand Prix, Circuit of Ardennes, and Vanderbilt Cup, the results are:—Darracq, once first, once second; Dietrich, once first; Renault, once first; F.I.A.T., twice second.

Those who are interested in the racing will find full reports in the various Motor Journals of the day.

Information sufficient to calculate the tractive forces, &c., is, naturally, seldom available. It may be of interest, however, to note that the Brazier car of 1904 had gear ratios of $3\frac{1}{2}$, 2, and $1\frac{1}{2}$ to 1, the corresponding tractive forces being about 470, 260, and 250 lbs. per ton, calculated as before, allowing that the car weighed about 24 cwts. when loaded.

Notes on Tables XI. and XII.

These give some particulars of the cars which ran in the Tourist Trophy race in the Isle of Man, in 1905 and 1906; the former being the first year it was run.

The essential condition of this race is that it is run with no limit to engine size, but with a limit of petrol for the course. In 1905 this was at the rate of 22·5 miles to the gallon, and in 1906 25 miles per gallon. There are certain other restrictions, as, for instance, the cars may not have more

than four speeds, chassis must not weigh less than 11·4 cwts., and the car must carry a total load of approximately 10 cwts. There are also some limits as to track, shape of body, &c., but the essential problem is to get a certain load round the course at the greatest possible speed with a given amount of petrol. This is a very complicated problem and only a few points can be gone into here.

It will be seen that in 1906 the car which won was very light indeed, being the lightest car which started except four, and that it had the largest total cylinder capacity and considerably the largest capacity per cwt. of any car which finished. Naturally, the smaller the weight to be moved the greater speed should be obtained as long as the car is strong enough to get round the course. One would also expect that a large engine running slowly would be more economical than a small one running fast, and this appears to be the case. Two ignitions should be a distinct advantage, and the winning car had two, placed at opposite sides of the cylinder.

Theoretically, it should be an advantage to have two large cylinders instead of four small ones, and the results of the race support this. In 1905 the race was won by a two-cylinder car with a considerable amount of petrol in hand. This year only one two-cylinder car started, but it secured seventh place, which is a far better performance than the average obtained by those with four cylinders. Further, it suffered from a nail puncture; otherwise it would probably have been second. It had only one ignition and a considerably smaller cylinder capacity than the winning car, so it seems as if with equal cylinder capacity and ignition a two-cylinder car should win easily. The difficulty would probably be to keep the weight down, but this might be managed.

In order to get the best results from an engine we should expect that it should be run at constant revolutions, and, therefore, many speeds would be required. Accordingly we find that all the successful cars had four, which is the limit allowed. In the winning car (1906) the ratios of gear are said to be 6·4, 3·7, 2·6, and 1·9 to 1, the wheels being 32 inches. This would give tractive forces of about 365, 225, 195, and 110 lbs. per ton on the various speeds, calculated in the usual way. The ratio of the top to bottom speed is 3·4 to 1. It will be seen on comparing these with those on Table viii. that the tractive forces on all the speeds are much less than those which most makers consider desirable for practical work, while the ratio between the top and bottom speeds is also very low. This seems right, for if we want to keep the engine to a constant low speed it is necessary to have a very high gear and the speeds as near together as they can be, consistent with the car going up the steepest hill on the route. The Arrol-Johnston and some of the most successful of the other cars are said to have been geared much the same.

It is possible the governor is not of much importance in a race, but if one is used it seems it should be of the cut-out type as in the Arrol-Johnston.

In fact, to design a special car suitable for this race it seems that the conditions of stationary gas engine practice should be studied, combined with that of the racing cars of the ordinary type.

Notes on Tables XIII. and XIV.

Tables xiii. and xiv. give some constructional features of the chassis exhibited, both at the motor show at Olympia, London, and also at the

Stanley Show at the Agricultural Hall, London, in 1905 and 1906 respectively.

The first divisions of Table xiv. show the simplicity or otherwise of the arrangement of the frame. Naturally the simplest is the straight frame with no inside frame; next come frames with no inside frame, but narrowed in front; then those both narrowed and raised; and last, these variations with an inside frame to carry the engine and gear box. In some cases there is an inside frame to carry the engine or gear box only.

It will be seen that the most usual construction is to have the frame narrowed in front, but with no inside frame and not raised. At the same time several of the best known makes have the frame so nearly straight that it is quite clear the modification can be of no practical advantage, while the bend is just sufficient to materially weaken the structure.

The next columns give the material, pressed steel being nearly universal.

The next columns give particulars of the springs. The single springs for both back and front springs are the most numerous, but a considerable number of makers fit either cross back or extension back springs to get greater effective length.

The next give particulars of the drive, the clutch, and the number and arrangement of speeds.

The next give the drive of the back shaft in the gear box where there is a direct drive. If the back shaft is driven at the front end the sliding gears are on the driven shaft; if it is driven at the back end the sliding gears are on the driving shaft. The advantages and disadvantages of these arrangements are referred to in Chapter xiii.

Next we have the number of universal joints between the gear box and back axle in the case of live-axle cars. Where there is only one joint the shaft runs in a tube forming the radius rod.

Brakes and radiators need no explanation. The next gives some idea of the number of gear wheels and countershafts employed on the engine to drive various cam shafts and other parts. It is not strictly comparative—first, because in some cases it is not easy to see exactly how many wheels there are in an engine; secondly, because in some cases lubricators, &c., are attached to the engine and driven by gear wheels, and in others these are on the dashboard and are driven by belts or worked by hand. Chain wheels for driving engine parts are included, but not belts or fan drives. The table also shows whether the gear wheels are enclosed in the crank case or are exposed outside; and where some are enclosed and some not the entries are put under the heading of "some enclosed."

Ignition LTM means low-tension magneto; HTM, high-tension magneto; HT, high tension with accumulators. It is difficult to separate the cars which are fitted with magneto only from those with high tension as well, as the majority of the cars with magneto are so arranged that high tension with accumulators can be fitted in addition, and in some cases they are generally so fitted though the chassis did not have the batteries, &c., there.

The columns as to pumps separate the centrifugal from the various forms of positive pumps, all the latter being classed as "gear," as this is the most common form. They are in all cases driven by gear wheels on the engine, except where specified as "friction driven." Where "thermosyphon" is specified there is no pump.

The next columns show whether the cylinders are cast in parts, all separate or all together.

The next columns give the arrangement of the valves in the engines.

Table xiii. gives a great many of the above particulars for the shows of 1905, and by comparison it can be seen in what direction particular makers are tending.

The tables are, however, not a very good index of this, and are no real index of the popularity of the different constructional features, for they only refer to the *chassis* exhibited, and few makers exhibited more than one chassis, which only represented the constructional features of that particular powered car. Also where makers exhibited several chassis of different powers, but all having the same features, these are not put in as separate chassis to save space. To get any idea of the popularity of the different chassis one ought to know how many cars of each are sold in a given time. Besides this many makers did not show a chassis at all, so that there are no particulars of their cars, though they may have sold a great many. This particularly affects the number of cars of three and four speeds, as where a maker makes his large cars with four speeds and his smaller ones with three, as is common, he usually exhibits the large chassis, though he may sell far more of the small cars.

In comparing the general tendencies of the 1906 show with those of the 1905 there is little difference to note in most points. There is a general tendency to increase the weight and this has been previously referred to. The live axle is coming into more general use, and in the general details design is being greatly improved; the cars and engines showing much more care in design, and not having various parts fixed to the frame and driven by belts, friction wheels, &c. The H-section axle has become very common, and is generally made with the jaws for the steering pivot also H-section. In this case the weight of the jaws is considerably less than that of those of the forged axle, as the jaws are made no stronger than the middle part of the axle, but the centre part where the greatest strain comes is generally at least as heavy as that of the forged axles. This has been referred to in Chapter xiv. At present, in fact, firms do not seem to trust an H-section axle to carry as high a stress per square inch as those which are tubular or solid; one firm of great reputation puts in an axle of such a size that the stress on it can hardly exceed 4,000 lbs. per square inch, though this is an exceptionally heavy one.

Several firms have taken up the casting of all the cylinders together, though Thornycroft and Renault, who showed engines made this way last year, are this year showing larger sized chassis in which the cylinders are separate.

The use of cross back springs has very greatly increased, and also of extension back springs, like that represented by fig. 275. This shows that the disadvantages of the short springs with which cars used to be fitted are getting to be generally recognised, and in all the better class cars where single springs are used these are of good length. There is still room for great improvement in some of the cheaper cars, which are fitted with side springs much too short for easy riding. The extra cost of springs of proper length is so small that there is no excuse for this. The double elliptic has not come into as great use as might have been expected, considering its advantages.

In some cases the extension spring is very neatly fitted into the

flange of the pressed steel frame, instead of being carried above it, as in fig. 275.

In clutches the cone still holds its own for moderate powers and is often used even for large powers, the two racing Darracqs being fitted with it. The disc clutch has, however, considerably increased in popularity.

Transmission is much as before, the smaller cars being fitted with three speeds, while the more powerful ones have four, which is obviously unnecessary. Several firms who fit four speeds still use the shaft-to-shaft drive, and this seems justified if it is considered that the four speeds are necessary. There is a slightly extended use of the Mors drive, combining a direct top with shaft-to-shaft for the other speeds. The "direct third, indirect fourth" has made no headway, and it seems in every way the best to have the direct drive on the top speed.

In the matter of engine design there are very much fewer cars built with the gear wheels exposed, and there is a great improvement in general design in the way of fewer pieces being bolted to the frame and driven with chains, &c., but some of the engines might be a good deal simplified with advantage. Comparatively few firms use chains for driving any of the engine parts, though there does not seem any very great objection to it, and one firm drives its engine parts this way. Chains cannot be conveniently enclosed, however, and gear drives are on the whole preferable. In some cases they are used to put the ignition apparatus on the dashboard; but, even in this case, gear can be used, and most firms prefer to keep the dashboard free.

Several of the large cars are fitted with two foot brakes. This does not seem as good as one large one, which, for the same total surface, can be made lighter, stronger, and cheaper. One reason given is that it is a good thing to have a foot brake which is not connected with the clutch. This is certainly true, but a simpler plan is to have only one foot brake and have the clutch free from any connection with brakes.

Several firms fit both brakes on the back hubs, one on the outside, the other on the inside, of the drums, instead of having one brake on the countershaft in the ordinary way and two on the back hubs. The advantage claimed for this is that the "braking" strain does not go through the differential gear, but as there is no difficulty in making the differential strong enough to take the braking strain this is no advantage. On the other hand, it has several disadvantages. It is much more expensive, as the foot brake necessitates two sets of brake blocks, &c., instead of one, and a compensating gear to make them act evenly; it will, therefore, be more than twice as expensive. Even then there is no compensating device so good as the ordinary differential. As one set of brakes is external this cannot be enclosed. As both brakes act on the same drum, if this gets hot there is no cool brake in reserve. The brake blocks cannot be made interchangeable.

In some chain-driven cars the foot brake acts on two brakes at the ends of the cross shaft, and is thus open to the same objections, except as regards heating.

In the live-axle cars much more attention is being paid to making the back axles accessible. They are often made like that shown in fig. 264, and so arranged that the top half of the casing can be taken off without disturbing any other part of the gear.

One feature of the 1906 shows was the number of six-cylinder cars exhibited. Whether the six-cylinder car will become at all general remains

to be seen, but its advantages do not seem to be sufficient to compensate for the extra complication. A study of Chapter iii. will show that the turning moment is rendered more even by increasing the number of cylinders, but the gain per cylinder lessens with every addition that is made.

Of the vertical engines of the ordinary type, with the cylinders in line, the four-cylinder is the smallest number in which the moving parts are balanced and the impulses evenly divided, while the unevenness of the turning moment is so small that, with a reasonable size of flywheel, they are not noticeable. Hence the margin for improvement by adding cylinders is very small.

The six-cylinder engine, on the other hand, has many disadvantages. In the first place, although theoretically it should be lighter for its power than the four-cylinder, in practice it does not seem to be so. This may be due to the difficulty of getting all the cylinders to work evenly, as it is noticeable that, while several short-distance records are held by cars with more than four cylinders, all the makers of long-distance racing cars prefer four. That is to say, that for an equal weight the multicylinder car can be made to give more power for a very short time when specially tuned up, but cannot be trusted to go on doing so even for a few hours. Racing experience is not, of course, conclusive, but in the simple matter of getting power much may be learnt from it.

The disadvantages of the six-cylinder engine are—its increased complication, the large proportion of room occupied by it, the inconvenient length of the wheel base, the weakness and expensiveness of the crank shaft, which must be machined from several centres, and the noise from numerous small parts. It would probably be better to make all engines diagonal when there are more than four cylinders to save length.

By doing this, the length and weight of the crank shaft, &c., are very much reduced as well as the space taken up in the car. To get even division of the impulses the cylinders should be arranged so as to form angles of 120° in the six-cylinder and of 90° for the eight-cylinder engine.

Up to date, according to my experience the best four-cylinder engines certainly run more quietly than the best six-cylinder, and as there does not seem to be any more vibration, there is no apparent advantage in having extra cylinders, but if it is desired to have more than four it might be best to go to the eight-cylinder diagonal at once. One firm last year was pushing such an engine but now shows only four- and six-cylinder cars.

TABLE XIII.

Construction of Cars Exhibited at the Motor Show, Olympia, and Stanley Show, Agricultural Hall, London, in 1905.

FRAMES.

Frame straight. No Inside Frame.	*Frame narrowed. No Inside Frame.*	*Frame narrowed, raised at back. No Inside Frame.*	*Frame straight. Inside Frame.*
Argyll.	Albion.	Arrol-Johnston.	Aster.
Bell.	Beaufort.	Brotherhood.	Chenard & Walcker.
Bollée.	Berliet.	Germain.	Climax.
Brooke.	Bianchi.	Renault.	Enfield.
80 h.p. Darracq.	De Dion.	Rochet-Schneider.	Horbick.
Iden.	De Dietrich.	Siddeley.	10 h.p. Humber.
James & Browne.	Dixi.		Maudslay.
Legros & Knowles.	F.I.A.T.		Mobile.
Rover.	Gobron-Brillié.		Panhard.
Siddeley.	M.A.B.		Simms.
Singer.	Mascot.		Standard.
Vulcan.	Mathieu.		Star.
	Mercedes.		Swift.
	Mors.		Vauxhall.
	Napier.		
	Peugeot.		
	Pilain.		
	Pipe.		
	Sunbeam.		

Frame straight, raised. Inside Frame.	*Inside Frame to Engine only.*	*Inside Frame to Gear Box only.*
14 C.G.V.	20 C.G.V.	Clément.
12 h.p. Talbot.	Crawshay-Williams.	Delaunay Belleville.
	Martini.	Gladiator.
	60 h.p. Panhard.	Minerva.
	Vauxhall.	Mobile.

FRAMES.			
Frame narrowed. Inside Frame.	*Frame narrowed and raised. Inside Frame.*	*Pressed Steel.*	*Wood and Steel.*
Ariel.	20 h.p. Darracq.	Argyll.	Aster.
Aster.	Delaunay Belleville.	Ariel.	Chenard & Walcker.
Belsize.		Arrol-Johnston.	Daimler.
Clément.		Beaufort.	Mobile.
Crawshay-Williams.		Bell.	Panhard.
Decauville.		Belsize.	Rover.
Dennis.		Berliet.	Simms.
Elswick.		Bianchi.	Spyker.
Gladiator.		Bollée.	Star.
16 h.p. Humber.		Brooke.	
Martini.		Brotherhood.	
Minerva.		Clément.	
Mobile.		Climax.	
Oldsmobile.		Crawshay-Williams.	
Rolls-Royce.		Darracq.	
25 h.p. Talbot.		Decauville.	
Thornycroft.		Delaunay Belleville.	
		Dennis.	*Tube.*
		De Dietrich.	
		Dixi.	
		Elswick.	C.G.V.
		Enfield.	10 h.p. Humber.
		F.I.A.T.	Maudslay.
		Germain.	
		Gladiator.	
		Gobron-Brillié.	
		Horbick.	
		16 h.p. Humber.	
		Iden.	
		24 h.p. James & Browne.	
		Legros & Knowles.	
		M.A.B.	*Channel Steel.*
		Martini.	
		Mascot.	
		Mathieu.	Albion.
		Mercedes.	18 h.p. James & Browne.
		Minerva.	
		Mobile.	
		Mors.	
		Napier.	
		National.	
		Oldsmobile.	
		Peugeot.	
		Pilain.	
		Pipe.	*I Section Steel.*
		Renault.	
		Rochet-Schneider.	
		Siddeley.	Singer.
		Standard.	
		Sunbeam.	
		Swift.	
		12 h.p. Talbot.	
		35 ,, ,,	
		Thornycroft.	
		Vulcan.	

DRIVE.		TRANSMISSION.	
Live Axle.	*Chains.*	*Three Speeds, run through.*	*Four Speeds, Gate.*
Argyll.	Albion.	10 h.p. Aster.	Ariel.
Ariel.	14 h.p. Aster.	12 ,, ,,	Arrol-Johnston.
Arrol-Johnston.	20 ,, ,,	20 ,, ,,	Bell.
10 h.p. Aster.	Bell.	Beaufort.	Berliet.
12 ,, ,,	Berliet.	Chenard & Walcker.	Bianchi.
Beaufort.	Bianchi.	Darracq.	Bollée.
Belsize.	Bollée.	Decauville.	Clément.
14 h.p. C.G.V.	Brooke.	Dennis.	Climax.
Clément.	Brotherhood.	Dixi.	Crawshay-Williams.
Climax.	20 h.p. C.G.V.	Enfield.	Daimler.
Darracq.	Crawshay-Williams.	Germain.	F.I.A.T.
Decauville.	Daimler.	10 h.p. Humber.	Gladiator.
Delaunay Belleville.	F.I.A.T.	Iden.	16 h.p. Humber.
Dennis.	Gladiator.	Minerva.	Martini.
Dixi.	Gobron-Brillié.	Mobile.	Maudslay.
Elswick.	James & Browne.	Napier.	Mercedes.
Enfield.	Martini.	Rover.	Rolls-Royce.
Germain.	Maudslay.	Simms.	Siddeley.
Horbick.	Mercedes.	Star.	Standard.
Hotchkiss.	Mors.	Swift.	Sunbeam.
Humber.	Napier.	9 h.p. Vauxhall.	Swift.
Iden.	Panhard.	10 h.p. Vulcan.	35 h.p. Talbot.
Legros & Knowles.	Peugeot.		Vauxhall.
M.A.B.	Pipe.		
Mascot.	Siddeley.		
Mathieu.	Star.		*Four, run through.*
Minerva.	Sunbeam.		
Mobile.	36 h.p. Talbot.		C.G.V.
Napier.	Vauxhall.		Delaunay Belleville.
National.			Mors.
Oldsmobile.			Panhard.
Renault.			Pipe.
Rochet-Schneider.			12 h.p. Talbot.
Rolls-Royce.			
Rover.			
Siddeley.	*Special.*	*Three Speeds, Gate.*	*Three with Cams.*
Simms.			
Spyker.			14 H.P. Aster.
Standard.	Chenard & Walcker.	Albion.	
Swift.	De Dion.	Argyll.	
12 h.p. Talbot.	Pilain.	Brooke.	*Three Special.*
Thornycroft.		Gobron-Brillié.	
Vulcan.		Horbick.	Renault.
Winton.		Legros & Knowles.	
		Mascot.	
		Mathieu.	*Four with Cams.*
		Oldsmobile.	
		Rochet-Schneider.	C.G.V.
		Siddeley.	M.A.B.
		Thornycroft.	
		20 h.p. Vulcan.	
			Separate Reverse Lever.
			Argyll.
			Daimler.

RADIATORS.		**GEAR WHEELS IN ENGINE.**		
Gilled.	*Honeycomb.*	*Enclosed.*	*Exposed.*	*Number.*
Albion.	Argyll.	Albion.	Argyll.	*Five.*
Arrol-Johnston.	Ariel.	Ariel.	12 h.p. Aster.	In most cases.
Beaufort.	Bell.	Arrol-Johnston.	14 ,, ,,	*Three.*
Brotherhood.	Belsize.	10 h.p. Aster.	20 ,, ,,	10 h.p. Aster.
C.G.V.	Berliet.	Beaufort.	Bell.	80 h.p. Darracq.
Daimler.	Bianchi.	Belsize.	Brotherhood.	Dixi.
Darracq.	Brooke.	Berliet.	Climax.	*Four.*
Delaunay Belleville.	Chenard & Walcker.	Bianchi.	Daimler.	Crawshay-Williams.
Gobron-Brillié.	Clément.	Brooke.	Dennis.	Darracq.
Humber.	Crawshay-Williams.	C.G.V.	Elswick.	Horbick.
Iden.	Decauville.	Chenard & Walcker.	F.I.A.T.	Renault.
Legros & Knowles.	De Dion.	Clément.	Horbick.	Spyker.
Minerva.	Dennis.	Crawshay-Williams.	Hotchkiss.	*Six.*
Mobile.	Elswick.	Darracq.	Legros & Knowles.	Beaufort.
Mors.	Enfield.	Decauville.	Martini.	Bollée.
Oldsmobile.	F.I.A.T.	De Dietrich.	Mascot.	Brooke.
Renault.	Germain.	De Dion.	Mercedes.	Clément.
Rolls-Royce.	Gladiator.	Delaunay Belleville.	Mobile.	Daimler.
Rover.	Horbick.	Dixi.	Oldsmobile.	De Dion.
Siddeley.	Hotchkiss.	Enfield.	Siddeley.	Gladiator.
Simms.	Mercedes.	F.I.A.T.	Simms.	Thornycroft.
Standard.	Martini.	Germain.	35 h.p. Talbot.	*Seven.*
Star.	Mascot.	Gladiator.	Vauxhall.	Decauville.
Vauxhall.	Mathieu.	Humber.		Belsize.
	Maudslay.	Minerva.		C.G.V.
	Napier.	Mors.		Germain.
	Panhard.	Napier.		*Eight.*
	Peugeot.	National.		Bell.
	Pilain.	Panhard.		Brotherhood.
	Spyker.	Pilain.		12 h.p. Talbot.
	Sunbeam.	Pipe.		*Nine.*
	12 h.p. Talbot	Renault.		C.G.V.
	35 ,, ,,	Rochet-Schneider.		*Ten.*
	Thornycroft.	Rolls-Royce.		Pipe.
	Vulcan.	Rover.		*Note.*—It is very difficult in many cases to see how many wheels there are exactly.
		Spyker.		
		Standard.		
		Sunbeam.		
		Swift.		
		12 h.p. Talbot.		
		Thornycroft.		
		Vulcan.		

AXLES.		PUMPS.	
Forged.	*Tube.*	*Gear, Gear-driven.*	*Centrifugal, Gear-driven.*
Albion.	Argyll.	Beaufort.	Albion.
14 h.p. Aster.	10 h.p. Aster.	Bell.	Argyll.
20 ,, ,,	Dennis.	Bollée.	12 h.p. Aster.
Beaufort.	Dixi.	C.G.V.	20 ,, ,,
Bell.	Horbick.	Crawshay-Williams.	Berliet.
Belsize.	Humber.	Daimler.	Bianchi.
Brooke.	Iden.	Darracq.	Brooke.
Brotherhood.	Legros & Knowles.	De Dietrich.	Brotherhood.
C.G.V.	Mathieu.	Elswick.	Clément.
Chenard & Walcker.	Oldsmobile.	Enfield.	Decauville.
Clément.	Renault.	Iden.	De Dion.
Climax.	Rover.	Legros & Knowles.	Delaunay Belleville.
Crawshay-Williams.	Simms.	Mathieu.	F.I.A.T.
Decauville.	Star.	Mors.	Gobron-Brillié.
Germain.	Spyker.	Oldsmobile.	Germain.
Gladiator.	Swift.	Rolls-Royce.	Gladiator.
Gobron-Brillié.	12 h.p. Talbot.	Rover.	Hotchkiss.
Hotchkiss.	Vulcan.	Siddeley.	Martini.
M.A.B.		Spyker.	Mercedes.
Mascot.		Thornycroft.	Panhard.
Mors.		Vulcan.	Pilain.
National.			Rochet-Schneider.
Panhard.			Simms.
Pilain.			12 h.p. Talbot.
Pipe.			35 ,, ,,
Sunbeam.			Vauxhall.
35 h.p. Talbot.	*I-Section.*	*Friction-driven.*	
Thornycroft.			
	Arrol-Johnston.	10 h.p. Aster.	
	12 h.p. Aster.	14 ,, ,,	
	Berliet.	Climax.	
	Bianchi.	Dennis.	
	Daimler.	Dixi.	
	De Dion.	Horbick.	
	Delaunay Belleville.	Mascot.	
	Elswick.	Mobile.	
	F.I.A.T.	Standard.	
	James & Browne.	Swift.	
	Martini.		
	Maudslay.		
	Mercedes.		
	Minerva.		
	Mobile.		
Cast.	Napier.	*No Pump.*	
	Peugeot.		
	Rochet-Schneider.		
Winton.	Rolls-Royce.	Chenard & Walcker.	
	Siddeley.	Belsize.	
	Singer.	Minerva.	
	Standard.	Renault.	
	Vauxhall.	Sunbeam.	

SPRINGS.
Cross Back.
Arrol-Johnston.
C.G.V.
De Dion.
Martini.
Mascot.
Mors.
Renault.
Rochet-Schneider.
Rolls-Royce.
Spyker.
24 h.p. Thornycroft.
Double Back.
Berliet.
Legros & Knowles.
20 h.p. Mercedes.
Sunbeam.
Extension Back.
Clément.
Dennis.
Gladiator.
Cross Front.
Rover.
All other cars single springs.

CYLINDERS.	
Pairs.	*Separate.*
Albion.	Argyll.
10 h.p. Aster.	Ariel.
Beaufort.	Arrol-Johnston.
Bell.	12 h.p. Aster.
Berliet.	14 ,, ,,
Bianchi.	20 ,, ,,
Bollée.	Belsize.
Brotherhood.	Brooke.
Chenard & Walcker.	Clément.
Crawshay-Williams.	Climax.
Daimler.	Delaunay Belleville.
Darracq.	Dennis.
Decauville.	Elswick.
Dixi.	Enfield.
F.I.A.T.	Germain.
Gobron-Brillié.	Gladiator.
Hotchkiss.	10 h.p. Humber.
Legros & Knowles.	16 ,, ,,
Martini.	Iden.
Mathieu.	James & Browne.
Maudslay.	Mascot.
Mercedes.	Mobile.
Minerva.	National.
Mors.	Panhard.
Napier.	16 h.p. Rover.
Oldsmobile.	Siddeley.
Peugeot.	Simms.
Pilain.	Standard.
Pipe.	Sunbeam.
Rochet-Schneider.	Swift.
Rolls-Royce.	12 h.p. Talbot.
Simms.	35 ,, ,,
Spyker.	24 h.p. Thornycroft.
Star.	Vauxhall.
Winton.	Vulcan.
All together.	
14 h.p. C.G.V.	
Renault.	
10 h.p. Rover.	
14 h.p. Thornycroft.	

IGNITION.

H.T.M.	*H.T. & H.T.M.*
Argyll.	Decauville.
Ariel.	De Dion.
Arrol-Johnston.	Gladiator.
20 h.p. Aster.	Gobron-Brillié.
Bell.	16 h.p. Humber.
Bollée.	Iden.
C.G.V.	Maudslay.
Chenard & Walcker.	Minerva.
Clément.	Pipe.
12 h.p. De Dietrich.	12 h.p. Talbot.
Dixi.	
Elswick.	
Enfield.	
Germain.	
Hotchkiss.	
Panhard.	
Renault.	
15 h.p. Siddeley.	
Simms.	
Sunbeam.	
Spyker.	
35 h.p. Talbot.	

L.T.M. & H.T.
20 h.p. Darracq.
Peugeot.
24 h.p. Thornycroft.

L.T.M.	*H.T.*
Albion.	10 h.p. Aster.
Arrol-Johnston.	12 ,, ,,
Beaufort.	14 ,, ,,
Belleville	Belsize.
Berliet.	Brotherhood.
Bianchi.	Climax.
Brooke.	Crawshay-Williams.
80 h.p. Darracq.	Daimler.
40 h.p. Dietrich.	Dennis.
F.I.A.T.	Horbick.
Martini.	10 h.p. Humber.
Mercedes.	Legros & Knowles.
Mors.	Mascot.
Pilain.	Mathieu.
Rochet-Schneider.	Mobile.
32 h.p. Siddeley.	National.
Simms.	Napier.
	Oldsmobile.
	Rolls-Royce.
	Rover.
	Standard.
	Star.
	Swift.
	14 h.p. Thornycroft.
	Vauxhall.

VALVES.

Opposite Sides.	*Same Side.*
Albion.	10 h.p. Aster.
Argyll.	C.G.V.
12 h.p. Aster.	Crawshay-Williams.
14 ,, ,,	Daimler.
20 ,, ,,	Darracq.
Beaufort.	Dixi.
Bell.	Horbick.
Berliet.	Iden.
Bianchi.	Legros & Knowles.
Bollée.	Napier.
Brotherhood.	Rover.
C.G.V.	Simms.
Chenard & Walcker.	Sunbeam.
Clément.	Swift.
Climax.	
Decauville.	
Delaunay Belleville.	
Dennis.	
Enfield.	
F.I.A.T.	
Germain,	
Gladiator.	
Gobron-Brillié.	
Hotchkiss.	
Humber.	
Martini.	
Mascot.	
Mathieu.	
Mercedes.	
Minerva.	
Mobile.	
Mors.	
National.	
Panhard.	
Pilain.	
Rochet-Schneider.	
Spyker.	
Standard.	
Swift.	
12 h.p. Talbot.	
35 ,, ,,	
24 h.p. Thornycroft.	
Vauxhall.	
Vulcan.	

Over each other.
Ariel.
Arrol-Johnston.
De Dion.
Elswick.
Napier.
Oldsmobile.
Renault.
Siddeley.
Simms.
7 h.p. Star.

Inlet on Top Exhaust Side.
Rolls-Royce.
10 h.p. Star.

Automatic Valves.
De Dion.
James & Browne.
Oldsmobile.
Renault.
7 h.p. Star.

Inclined.	*All on Top.*
Daimler.	Belsize.
Pipe.	80 h.p. Darracq.
	Maudslay.
	14 h.p. Thornycroft.

<table>
<tr><th colspan="2">BRAKES.</th><th colspan="2">CLUTCHES.</th></tr>
<tr><td>External Foot.
Internal Back.</td><td>Both External.</td><td>Cone.</td><td>Inverted Cone.</td></tr>
<tr><td rowspan="5">Argyll.
Ariel.
10 h.p. Aster.
12 ,, ,,
20 ,, ,,
Bell.
Berliet.
Bianchi.
Bollée.
Brotherhood.
Clément.
Crawshay-Williams.
Decauville.
De Dietrich.
De Dion.
Dixi.
Elswick.
F.I.A.T.
Germain.
Gladiator.
Gobron-Brillié.
Horbick.
Hotchkiss.
Iden.
James & Browne.
Martini.
Mascot.
Mercedes.
Minerva.
Mors.
Napier,
Oldsmobile.
Panhard.
Peugeot.
Pilain.
Rochet-Schneider.
Rolls-Royce.
Siddeley.
Simms.
Star.
Talbot.
Vauxhall.
Vulcan.</td><td>Albion.
Delaunay Belleville.
Enfield.
10 h.p. Humber.
Mobile.
Standard.
Swift.
Thornycroft.
10 h.p. Vulcan.</td><td rowspan="3">Albion.
Bollée.
Brooke.
Brotherhood.
Chenard & Walcker.
Daimler.
Darracq.
Delaunay Belleville.
Dennis.
Dixi.
Elswick.
Enfield.
Germain.
Hotchkiss.
Humber.
Mascot.
Maudslay.
Minerva.
Mobile.
Napier.
Oldsmobile.
Peugeot.
Pilain.
Siddeley.
Simms.
Spyker.
Star.
Swift.
12 h.p. Talbot.
35 ,, ,,
Vauxhall.
Vulcan.</td><td>Ariel.
Aster.
Beaufort.
Decauville.
Horbick.
Iden
Mathieu.
Renault.
Rolls-Royce.
Sunbeam.</td></tr>
<tr><td>Both Internal.</td><td>Expanding.</td></tr>
<tr><td>Arrol-Johnston.
14 h.p. Aster.
Beaufort.
Brooke.
C.G.V.
Climax.
20 h.p. Darracq.
16 h.p. Humber.
Spyker.
35 h.p. Talbot.
Vauxhall.</td><td>Berliet.
Bianchi.
Climax.
Martini.
Mercedes.
Mors.
Rochet-Schneider.
Rover.</td></tr>
<tr><td>Brakes not through Differential.</td><td>Special.</td><td>Disc.</td></tr>
<tr><td>Belsize.
Brotherhood.
Daimler
Legros & Knowles.
Maudslay.</td><td>De Dion.
Gobron-Brillié.</td><td>Argyll.
Belsize.
Clément.
Crawshay-Williams.
F.I.A.T.
Gladiator.
Legros & Knowles.
60 h.p. Panhard.
Standard.
Thornycroft.</td></tr>
</table>

TABLE XIV.

Construction of the Cars Exhibited at the Motor Show at Olympia, and the Stanley Show at Agricultural Hall in 1906.

FRAMES.			
Frame straight. No Inside Frame.	*Frame narrowed. No Inside Frame.*	*Frame narrowed, raised at back. No Inside Frame.*	*Frame straight. With Inside Frame.*
Argyll.	Adams.	Crossley.	Airex.
Arrol-Johnston.	Albion.	14 F.I.A.T.	Alldays.
Cadillac.	Armstrong.	10 Gregoire.	Brooke.
C.G.V.	Arrol-Johnston.	Hotchkiss.	Humber.
Daimler.	Beaufort.	Metallurgique.	Laurin & Klement.
80 Darracq.	Belleville.	15 Mors.	Maudslay.
Enfield.	Belsize.	Nordenfeldt.	Panhard.
Ford.	Bentall.	Rochet-Schneider.	Spyker.
Maxwell.	Benz.	S.C.A.R.	10 Star.
Napier.	Berliet.	15 Siddeley.	Stuart.
Simms.	Bianchi.		Swift.
Winton.	Britannia.		Vauxhall.
	Brotherhood.		
	Brown.		
	C.S.B.		
	Deasy.		
	De Dietrich.		
	De la Buire.		
	Delahaye.		
	30 F.I.A.T.		
	60 ,,		
	Florentia.		
	Germain.		
	Gladiator.		
	Gobron-Brillié.		
	Iris.		
	Isotta-Fraschini.		
	Itala.		
	James & Browne.		
	Junior.		
	Mascot.		
	Mercedes.		
	Mieusset.		
	Morgan.		
	28 Mors.		
	Nagant.		
	Pilain.		
	Radia.		
	Rapid.		
	Renault.		
	Rolls-Royce.		
	Seymour-Turner.		
	45 Siddeley.		
	Speedwell.		
	Sunbeam.		
	Vulpes		
	Zust.		

FRAMES.			
Frame narrowed. With Inside Frame.	*Frame narrowed and raised. With Inside Frame.*	*Pressed Steel.*	*Wood and Steel.*
Ariel. Chenard & Walcker. Clément. 7 Cupelle. Dennis. Marlboro'. 30 Standard. 16 Star. 30 ,, Thornycroft. Vinot.	Aster. Austin. 20 Darracq. De la Buire. Martini. Minerva. 30 Siddeley. Talbot. West.	Adams. Airex. Alldays. Argyll. Ariel. Armstrong. Arrol-Johnston. Aster. Austin. Beaufort. Bell. Belleville. Belsize. Bentall. Benz. Berliet. Bianchi. Bronhot. Britannia. Brotherhood. Brooke. Brown. Cadillac. Chenard & Walcker. Clément. Climax. Courier. Crossley. C.S.B. 7 Cupelle. Daimler. Darracq. De Dietrich. De Dion. De la Buire. Delahaye. Dennis. Enfield. F.I.A.T. Florentia. Ford. Germain. Gladiator. Gobron-Brillié. Gregoire. Hotchkiss. Humber. Iris. Isotta-Fraschini. Itala. James & Browne. Junior. Laurin & Klement. Leader. Lindsay. Marlboro'. Martini. Mascot. Mass. Maxwell. Mayfair. Mercedes. Metallurgique. Mieusset. Minerva. Morgan. Mors. Nagant. Napier. Nordenfeldt. Pilain. Radia. Rapid. Renault. Robinson-Hole. Rochet-Schneider. Rolls-Royce. S.C.A.R. Seymour-Turner. Siddeley. Speedwell. Standard. Star. Swift. Talbot. Thornycroft. Vauxhall. 16 Vinot. 35 ,, Vulpes. West. Winton. Zust.	Cupelle. Deasy. Panhard. Simms. Spyker. Stuart. Sunbeam.
Narrowed. Inside Frame to Engine only.			*Tube.*
Bronhot. De Dion. Mass.			C.G.V. Maudslay.
Inside Frame to Gear Box only.			*Channel.*
Bell. Courier. 12 Leader. Lindsay. Mayfair. Minerva. 50 Standard.			Albion.

SPRINGS.				
Single.		*Cross Back.*	*Extension Back.*	*Double Back.*
Airex.	Stuart.	24 Arrol-Johnston.	Adams.	24 Albion.
12 Albion.	Swift.	12 Aster.	Armstrong.	Bentall.
Alldays.	Vauxhall.	Belleville.	Beaufort.	Sunbeam.
Argyll.	Vulpes.	Britannia.	Cadillac.	
Ariel.	Winton.	Brown.	Clément.	
12 Arrol-Johnston.	Zust.	C.G.V.	Crossley.	
20 Aster.		Chenard & Walcker.	Dennis.	
24 ,,		7 Cupelle.	18 Florentia.	
Austin.		De Dion.	Gladiator.	
Bell.		De la Buire.	Isotta-Fraschini.	
Belsize.		Delahaye.	Junior.	
Benz.		25 Enfield.	15 Mors.	
Berliet.		10 Gregoire.	Panhard.	
Bianchi.		James & Browne.	Pilain.	
Bronhot.		Marlboro'.	S.B.C.	
Brooke.		Martini.	30 Siddeley.	
Brotherhood.		Mascot.	Speedwell.	
Climax.		Mass.	16 Vinot.	*Double Back. Cross Front.*
Courier.		Mayfair.	35 ,,	
Cupelle.		Mieusset.		Ford.
Daimler.		Minerva.		
Darracq.		28 Mors.		
De Dietrich.		Napier.		
24 De la Buire.		Nordenfeldt.		
15 Enfield.		Renault.		
F.I.A.T.		Rochet-Schneider.		
40 Florentia.		Rolls-Royce.		
Germain.		S.C.A.R.		
Gobron-Brillié.		Simms.		
25 Hotchkiss.		Spyker.		
45 ,,		Talbot.		
Humber.		Thornycroft.		
Iris.		West.		
Itala.				
Laurin & Klement.				
Leader.				*Single Back. Cross Front.*
Lindsay.				
Maudslay.				Deasy.
Maxwell.				
Mercedes.				
Metallurgique.				
24 Minerva.				
Morgan.				
Nagant.				
Radia.				
Rapid.				
Robinson & Hole.				
Seymour-Turner.				
15 Siddeley.				
45 ,,				
Star.				
Standard.				

AXLE.			
H.		*Solid.*	*Tube.*
Adams.	Siddeley.	Albion.	Airex.
Ariel.	Simms.	Aster.	Alldays.
Arrol-Johnston.	Speedwell.	Bronhot.	Argyll.
Austin.	Standard.	Chenard & Walcker.	Armstrong.
Beaufort.	Star.	Delahaye.	Cadillac.
Bell.	Talbot.	Hotchkiss.	7 Cupelle.
Belleville.	Thornycroft.	James & Browne.	Deasy.
Belsize.	Vauxhall.	Leader.	Dennis.
Bentall.	35 Vinot.	Mass.	Humber.
Benz.	West.	Mieusset.	Marlboro'.
Berliet.	Zust.	Morgan.	Maxwell.
Bianchi.		28 Mors.	Renault.
Britannia.		Panhard.	Robinson & Hole.
Brooke.		Radia.	S.B.C.
Brotherhood.		Sunbeam.	Spyker.
Brown.		16 Vinot.	Stuart.
C.G.V.			Swift.
Clément.			Vulpes.
Climax.			
Courier.			
Crossley.			
Cupelle.			
Daimler.			
Darracq.			
De Dietrich.			
De Dion.			
De la Buire.			
Enfield.			
F.I.A.T.			
Florentia.			
Ford.			
Germain.			
Gladiator.			
Gobron-Brillié.			
16 Gregoire.			
Iris.			
Isotta-Fraschini.			*Cast.*
Itala.			
Junior.			
Lindsay.			Winton.
Martini.			
Mascot.			
Maudslay.			
Mayfair.			
Mercedes.			
Metallurgique.			
Minerva.			
15 Mors.			
Nagant.			
Napier.			
Nordenfeldt.			
Pilain.			
Rapid.			
Rochet-Schneider.			
Rolls-Royce.			
S.C.A.R.			
Seymour-Turner.			

DRIVE.			
Live Axle.		*Side Chain.*	*Special Drive.*
Adams.	S.C.A.R.	Albion.	De Dion.
Airex.	Seymour-Turner.	20 Aster.	Chenard & Walcker.
Alldays.	15 Siddeley.	24 ,,	Pilain.
Argyll.	20 ,,	25 Austin.	
Ariel.	Simms.	Beaufort	
Armstrong.	Speedwell.	30 Bell.	
Arrol-Johnston.	Spyker.	Belleville.	
12 Aster.	Standard.	Berliet.	
15 Austin.	10 Star.	Bianchi.	
16 Bell.	Stnart.	Bronhot.	
20 ,,	Swift.	Brotherhood.	
Belsize.	Talbot.	C.G.V.	
Bentall.	Thornycroft.	Clément.	
Benz.	Vauxhall.	Daimler.	
Britannia.	Vulpes.	De Dietrich.	
Brooke	West.	De la Buire.	
Brown.	Winton.	Delahaye.	
Cadillac.		30 F.I.A.T.	
Climax.		60 ,,	
Courier.		40 Florentia.	
Crossley.		Gladiator.	
Cupelle.		Gobron-Brillié.	
Darracq.		Isotta-Fraschini.	
Deasy.		James & Browne.	
20 De la Buire.		Junior.	
Dennis.		Maudslay.	
Enfield.		Mercedes.	
14 F.I.A.T.		Mieusset.	
18 Florentia.		28 Mors.	
Ford.		Nagant.	
Germain.		Panhard.	
Gregoire.		Radia.	
Hotchkiss.		45 Siddeley.	
Humber.		16 Star.	
Iris.		30 ,,	
Itala.		Sunbeam.	
Laurin & Klement.		Vinot.	
Leader.		Zust.	
Lindsay.			
Marlboro'.			
Martini.			
Mascot.			
Mass.			
Maxwell.			
Mayfair.			
Metallurgique.			
Minerva.			
Morgan.			
15 Mors.			
Napier.			
Nordenfeldt.			
Rapid.			
Renault.			
Robinson & Hole.			
Rochet-Schneider.			
Rolls-Royce.			
S.B.C.			

CLUTCH.

Disc.	*Cone.*	*Inverted Cone.*	*Expanding and Special.*	*Metal Cone.*
24 Albion.	Airex.	12 Aster.	Bronhot.	Bentall.
Argyll.	12 Albion.	Brown.	Crossley.	Climax.
Ariel.	Alldays.	Dennis.	De Dietrich.	16 James & Browne.
Armstrong.	20 Aster.	Minerva.	De Dion.	Napier.
Arrol-Johnston.	16 Bell.	Renault.	De la Buire.	30 Siddeley.
24 Aster.	Belleville.	Rolls-Royce.	Florentia.	45 ,,
Austin.	Benz.		Mascot.	Vauxhall.
Beaufort.	Britannia.		Mercedes.	
20 Bell.	Brotherhood.		Metallurgique.	
30 ,,	C.G.V.		Mors.	
Belsize.	Chenard & Walcker.		Pilain.	
Berliet.	Courier.		West.	
Bianchi.	C.S.B.			
Brooke.	Cupelle.			
Clément.	Daimler.			
Deasy.	Darracq.			
25 Enfield.	15 Enfield.			
F.I.A.T.	Germain.			
Gladiator.	Gobron-Brillié.			
Iris.	Gregoire.			
Isotta-Fraschini.	Hotchkiss.			
Itala.	Humber.			
30 James & Browne.	Leader.			
Junior.	Lindsay.			
16 Maxwell.	Marlboro'.			
Nagant.	Martini.			
Rapid.	Mascot.			
Seymour-Turner.	Mass.			
Speedwell.	Maudslay.			
Standard.	Mayfair.			
Swift.	Mieusset.			
Thornycroft.	Morgan.			
35 Vinot.	Nordenfeldt.			
Winton.	Panhard.			
Zust.	Radia.			
	Robinson & Hole.			
	Rochet-Schneider.			
	S.C.A.R.			
	15 Siddeley.			
	Simms.			
	Spyker.			
	Star.			
	Stuart.			
	Sunbeam.			
	Talbot.			
	16 Vinot.			
	Vulpes.			

TRANSMISSION.

Three Speeds. Run through.	*Three Speeds. Gate.*	*Four Speeds. Run through.*	*Four Speeds. Gate.*	*Shaft-to-Shaft Drive.*
Airex. Alldays. Belsize. Bentall. Britannia. Courier. Cupelle. Dennis. 15 Enfield. Germain. Gregoire. Humber. Laurin & Klement. Leader. Lindsay. Marlboro'. Mass. 16 Maxwell. S.C.A.R. Star. Stuart. Vulpes. West. *Two Speeds. Run through.* 80 Darracq. 200 ,, *Two Speeds. Crypto Gear.* Ford. 8 Maxwell.	12 Albion. Argyll. 16 Bell. 20 ,, C.S.B. 25 Enfield. Iris. Minerva. 15 Mors. Napier. Robinson & Hole. Rochet-Schneider. 15 Siddeley. Simms. Speedwell. Thornycroft. Vauxhall. 16 Vinot.	Armstrong. Belleville. C.G.V. Chenard & Walcker. Mayfair. 28 Mors. Nordenfeldt. Panhard. *Four, with Cams.* Deasy.	24 Albion. Ariel. Arrol-Johnston. Austin. Beaufort. 30 Bell. Benz. Berliet. Bianchi. Bronhot. Brooke. Brotherhood. Brown. Clément. Climax. Crossley. Daimler. 20 Darracq. De Dietrich. De Dion. De la Buire. Delahaye. F.I.A.T. Florentia. Gladiator. Gobron-Brillié. Hotchkiss. Isotta-Fraschini. Itala. James & Browne. Junior. Martini. Maudslay. Mercedes. Metallurgique. Mieusset. Morgan. Nagant. Pilain. Radia. Rapid. Rolls-Royce. Seymour-Turner. 30 Siddeley. 45 ,, Standard. Sunbeam. Swift. Talbot. 35 Vinot. Winton. Zust.	Austin. Beaufort. Bianchi. Brotherhood. Daimler. De Dietrich. 30 F.I.A.T. 60 ,, James & Browne. Panhard. *Mors Drive.* Mercedes. 28 Mors. Zust. *Direct Third. Indirect Fourth.* Climax. Rolls-Royce. Standard. Winton.

BACK SHAFT DRIVEN.		UNIVERSAL JOINTS.	
Front End.	*Back End.*	*One.*	*Two.*
12 Albion.	24 Albion.	Ariel.	Adams.
Alldays.	Belleville.	12 Aster.	Airex.
Argyll.	Belsize.	Belsize.	Alldays.
Ariel.	Bentall.	Bentall.	Argyll.
Armstrong.	Benz.	Benz.	Armstrong.
Arrol-Johnston.	Brooke.	Brooke.	Arrol-Johnston.
Aster.	C.G.V.	7 Cupelle.	Bell.
Austin.	Cupelle.	Deasy.	Britannia.
Bell.	Dennis.	F.I.A.T.	Brown.
Britannia.	F.I.A.T.	Florentia.	Cadillac.
Chenard & Walcker.	Germain.	Ford.	Chenard & Walcker.
Clément.	Gobron-Brillié.	Iris.	Climax.
Climax.	Hotchkiss.	Lindsay.	Courier.
Crossley.	Itala.	Minerva.	Crossley.
C.S.B.	Mascot.	Rolls-Royce.	C.S.B.
7 Cupelle.	Mass.	S.C.A.R.	Cupelle.
Darracq.	Mayfair.	Seymour-Turner.	Darracq.
Deasy.	Metallurgique.	Simms.	De la Buire.
De la Buire.	Mieusset.	Spyker.	Dennis.
Enfield.	Morgan.	Talbot.	Enfield.
Florentia.	15 Mors.	West.	Germain.
Gladiator.	Napier.		Gregoire.
Gregoire.	Radia.		Hotchkiss.
Iris.	S.C.A.R.		Itala.
Leader.	Simms.		Leader.
Marlboro'.	Speedwell.		Marlboro'.
Martini.	Star.		Martini.
Maudslay.	Stuart.		Mascot.
Minerva.			Mass.
Nagant.			Maxwell.
Nordenfeldt.			Mayfair.
Rapid.			Metallurgique.
Rochet-Schneider.			Morgan.
Rolls-Royce.			Napier.
Seymour-Turner.			Nordenfeldt.
Siddeley.			Rapid.
Standard.			Renault.
Swift.			Robinson & Hole.
Thornycroft.			Rochet-Schneider.
Vauxhall.			15 Siddeley.
Vinot.			30 ,,
West.			Speedwell.
			Standard.
			10 Star.
			Stuart.
			Swift.
			Thornycroft.
			Vauxhall.
			Vulpes.
			Winton.

BRAKES.				
Foot External. Side Internal.		*Both External.*	*Both Internal.*	*Both on Back Hub.*
Adams.	Rochet-Schneider.	Airex.	Alldays.	Cadillac.
24 Albion.	Rolls-Royce.	12 Albion.	Arrol-Johnston.	Climax.
Argyll.	S.C.A.R.	Chenard & Walcker.	Belsize.	Deasy.
Ariel.	Seymour-Turner.	Daimler.	Brooke.	10 Star.
Armstrong.	Siddeley.	Humber.	C.G.V.	Winton.
Aster.	Speedwell.	Vauxhall.	De la Buire.	
Austin.	Spyker.		Gregoire.	*Foot-brake on Differential. In Gear Case.*
Beaufort.	Standard.		Leader.	Bentall.
Bell.	16 Star.		Mass.	Sunbeam.
Belleville.	30 ,,		Maudslay.	
Benz.	Stuart.		Mieusset.	*Two Foot-brakes.*
Berliet.	Swift.		Morgan.	Armstrong.
Bianchi.	Thornycroft.		Nordenfeldt.	40 Bianchi.
Britannia.	Vulpes.		Radia.	70 ,,
Bronhot.	West.		Renault.	Bronhot.
Brotherhood.	Zust.		Robinson & Hole.	Brotherhood.
Brown.			Simms.	Clément.
Clément.			Talbot.	Crossley.
Courier.			Vinot.	De la Buire.
Crossley.				Gladiator.
C.S.B.				Gobron-Brillié.
Cupelle.				Isotta-Fraschini.
Darracq.				Junior.
De Dietrich.				Mercedes.
De Dion.				Metallurgique.
De la Buire.				Mieusset.
Delahaye.				Nagant.
Dennis.				Spyker.
Enfield.				
F.I.A.T.				
Florentia.				
Ford.				
Germain.				
Gladiator.				
Gobron-Brillié.				
Hotchkiss.				
Iris.				
Isotta-Fraschini.				
Itala.				
James & Browne.				
Junior.				
Lindsay.				
Marlboro'.				
Martini.				
Mascot.				
Maxwell.				
Mayfair.				
Mercedes.				
Metallurgique.				
Minerva.				
Mors.				
Nagant.				
Napier.				
Panhard.				
Pilain.				
Rapid.				

RADIATORS.			
Honeycomb.		*Gilled Tube.*	*Special.*
Argyll.	Swift.	Airex.	Cadillac.
Ariel.	Talbot.	Albion.	Maxwell.
Austin.	Thornycroft.	Alldays.	
Beaufort.	Vinot.	Armstrong.	
20 Bell.	Vulpes.	Arrol-Johnston.	
30 ,,	Zust.	Aster.	
Belleville.		16 Bell.	
Belsize.		Benz.	
Bentall.		Bronhot.	
Berliet.		C.G.V.	
Bianchi.		Climax.	
Britannia.		Cupelle.	
Brooke.		Daimler.	
Brotherhood.		Darracq.	
Brown.		Deasy.	
Chenard & Walcker.		De Dietrich.	
Clément.		De Dion.	
Courier.		Gobron-Brillié.	
Crossley.		Gregoire.	
C.S.B.		Itala.	
De la Buire.		16 James & Browne.	
Delahaye.		Laurin & Klement.	
Dennis.		Mayfair.	
Enfield.		Mieusset.	
F.I.A.T.		Minerva.	
Florentia.		Mors.	
Ford.		Nordenfeldt.	
Germain.		Panhard.	
Gladiator.		Renault.	
Hotchkiss.		Rochet-Schneider.	
Iris.		Rolls-Royce.	
Isotta-Fraschini.		Siddeley.	
30 James & Browne.		Standard.	
Junior.		10 Star.	
Leader.		Vauxhall.	
Lindsay.		West.	
Marlboro'.		Winton.	
Martini.			
Mascot.			
Mass.			
Maudslay.			
Mercedes.			
Metallurgique.			
Morgan.			
Nagant.			
Pilain.			
Radia.			
Rapid.			
S.C.A.R.			
Seymour-Turner.			
Simms.			
Speedwell.			
16 Star.			
30 ,,			
Stuart.			
Sunbeam.			

PUMPS.

Centrifugal.		*Gear.*	*Thermo-syphon.*
Albion.	Vinot.	Airex.	Belsize.
Alldays.	West.	Armstrong.	Chenard & Walcker.
Argyll.	Winton.	Arrol-Johnston.	7 Cupelle.
Ariel.	Zust.	Beaufort.	Gregoire.
Aster.		Benz.	Laurin & Klement.
Austin.		Britannia.	Marlboro'.
Bell.		Brown.	Maxwell.
Belleville.		C.G.V.	Minerva.
Bentall.		Crossley.	Renault.
Berliet.		Daimler.	S.C.A.R.
Bianchi.		Darracq.	Sunbeam.
Bronhot.		Deasy.	
Brooke.		Enfield.	
Brotherhood.		Gobron-Brillié.	
Cadillac.		Humber.	
Clément.		Iris.	
Climax.		Leader.	
Courier.		Maudslay.	
C.S.B.		Mayfair.	
Cupelle.		Morgan.	
De Dietrich.		Mors.	
De Dion.		Seymour-Turner.	
De la Buire.		Simms.	
Delahaye.		Spyker.	
Dennis.		Thornycroft.	
F.I.A.T.		Vulpes.	
Florentia.			
Germain.			
Gladiator.			
Hotchkiss.			
Isotta-Fraschini.			
Itala.			
James & Browne.			*Friction-driven Centrifugal.*
Junior.			
Lindsay.			
Martini.			
Mascot.			Stuart.
Mass.			
Mercedes.			
Metallurgique.			
Mieusset.			
Nagant.			
Napier.			
Panhard.			
Pilain.			
Radia.			
Rapid.			
Rochet-Schneider.			
Rolls-Royce.			
Siddeley.			
Speedwell.			
Standard.			
Star.			
Swift.			
Talbot.			
Vauxhall.			

CYLINDERS CAST.		
Pairs.	*Separate.*	*All together.*
Albion.	Adams.	C.S.B.
Alldays.	Airex.	Deasy.
Ariel.	Argyll.	Gregoire.
Beaufort.	Armstrong.	15 Mors.
Bell.	Arrol-Johnston.	
Belsize.	Aster.	
Benz.	Austin.	
Berliet.	Belleville.	
Bianchi.	Bentall.	
Britannia.	Brown.	
Bronhot.	Cadillac.	
Brooke.	C.G.V.	
Brotherhood.	Clément.	
Chenard & Walcker.	Climax.	
Courier.	De Dion.	
Crossley.	Delahaye.	
Cupelle.	Dennis.	
Daimler.	Germain.	
Darracq.	Gladiator.	
De Dietrich.	Humber.	
De la Buire.	James & Browne.	
Enfield.	Junior.	
F.I.A.T.	Laurin & Klement.	
Ford.	Leader.	
Gobron-Brillié.	Mascot.	
Hotchkiss.	Maxwell.	
Iris.	Morgan.	
Isotta-Fraschini.	28 Mors.	
Itala.	Nordenfeldt.	
Lindsay.	Panhard.	*In Threes.*
Martini.	Rapid.	
Mass.	S.C.A.R.	
Maudslay.	Seymour-Turner.	Rolls-Royce.
Mayfair.	15 Siddeley.	
Mercedes.	Standard.	
Metallurgique.	Sunbeam.	
Mieusset.	Swift.	
Minerva.	Talbot.	
Nagant.	Vauxhall.	
Napier.	West.	
Pilain.		
Radia.		
Renault.		
Robinson & Hole.		
Rochet-Schneider.		
Siddeley.		
Simms.		
Speedwell.		
Spyker.		
Star.		
Stuart.		
Thornycroft.		
16 Vinot.		
35 ,,		
Vulpes.		
Winton.		
Zust.		

VALVES.

Opposite.		*Same Side.*	*All at Top.*
Albion.	Talbot.	Adams.	Belsize.
Alldays.	Thornycroft.	16 Bell.	80 Darracq.
Argyll.	Vauxhall.	Bronhot.	200 ,,
Armstrong.	16 Vinot.	Brooke.	Maudslay.
Aster.	35 ,,	Cadillac.	
Austin.	Vulpes.	Courier.	
Beaufort.	West.	Cupelle.	
20 Bell.		Daimler.	
30 ,,		20 Darracq.	
Belleville.		60 ,,	
Bentall.		De Dietrich.	
Benz.		De Dion.	
Berliet.		Ford.	
Bianchi.		8 Gregoire.	
Britannia.		Iris.	
Brotherhood.		Lindsay.	
Brown.		Marlboro'.	
C.G.V.		Mass.	*Over each other.*
Chenard & Walcker.		Mayfair.	
Clément.		15 Mors.	
Climax.		Napier.	Airex.
Crossley.		Renault.	Ariel.
C.S.B.		Rolls-Royce.	Arrol-Johnston.
Deasy.		Seymour-Turner.	10 Gregoire.
De la Buire.		30 Siddeley.	Laurin & Klement.
Delahaye.		45 ,,	15 Siddeley.
Dennis.		Simms.	
Enfield.		Speedwell.	
F.I.A.T.		16 Star.	
Florentia.		30 ,,	
Germain.		Stuart.	
Gladiator.		Sunbeam.	
Hotchkiss.		Swift.	
Humber.		Winton.	
Isotta-Fraschini.		Zust.	
Itala.			
James & Browne.			
Junior.			
Leader.			
Martini.			
Mascot.			*Inlet at Top. Exhaust at Side.*
Mercedes.			
Mieusset.			
Minerva.			Metallurgique.
Morgan.			10 Star.
28 Mors.			
Nagant.			
Nordenfeldt.			
Panhard.			
Pilain.			
Radia.			
Rapid.			
Robinson & Hole.			
Rochet-Schneider.			
S.C.A.R.			
Spyker.			
Standard.			

GEAR WHEELS AND CHAIN WHEELS ON ENGINE.

Two.	*Five.*	*Six.*	*Eight.*
7 Cupelle.	Ariel.	Armstrong.	Bell.
Marlboro'.	Arrol-Johnston.	Brooke.	Brotherhood.
Maxwell.	15 Austin.	Courier.	C.G.V.
Stuart.	Beaufort.	Daimler.	De Dion.
	Belleville.	15 Enfield.	12 Leader.
	Bentall.	Humber.	Lindsay.
	Berliet.	Mass.	Maudslay.
	Britannia.	Mayfair.	Robinson & Hole.
	Cadillac.	Panhard.	Simms.
	Chenard & Walcker.	Standard.	Sunbeam.
	Clément.	Swift.	
	Crossley.	35 Vinot.	
	20 Darracq.		
	Deasy.		
	De Dietrich.		
	De la Buire.		
	F.I.A.T.		
Three.	Florentia.		*Nine.*
	Gladiator.		
	Gobron-Brillié.		
Adams.	Hotchkiss.		Belsize.
80 Darracq.	Isotta-Fraschini.		Cupelle.
Gregoire.	Itala.		Dennis.
10 Leader.	30 James & Browne.		25 Enfield.
	Junior.		28 Mors.
	Martini.		Napier.
	Mascot.		Vauxhall.
	Mercedes.		
	Metallurgique.		
	Mieusset.		
	24 Minerva.		
	15 Mors.		
	Nordenfeldt.		
	Pilain.		
	Radia.		
Four.	Rapid.	*Seven.*	*Eleven.*
	Rochet-Schneider.		
	S.C.A.R.		
Airex.	10 Star.	Argyll.	30 Siddeley.
12 Albion.	16 Vinot.	Aster.	
Alldays.	West.	25 Austin.	
Ford.		Benz.	
Laurin & Klement.		Bronhot.	
40 Minerva.		Brown.	
Renault.		C.S.B.	
Seymour-Turner.		60 Darracq.	
Spyker.		Delahaye.	
		Iris.	*Thirteen.*
		Morgan.	
		Nagant.	
		Rolls-Royce.	45 Siddeley.
		Speedwell.	
		16 Star.	
		30 ,,	
		Talbot.	
		Thornycroft.	

GEAR WHEELS AND CHAIN WHEELS ON ENGINE.			
Enclosed.		*Exposed.*	*Some Enclosed.*
Airex.	Rochet-Schneider.	Adams.	Ariel.
Albion.	Rolls-Royce.	Bell.	Bronhot.
Alldays.	S.C.A.R.	Daimler.	Brooke.
Argyll.	Simms.	Hotchkiss.	Brotherhood.
Armstrong.	Speedwell.	Mascot.	Cupelle.
Arrol-Johnston.	Spyker.	Mass.	Lindsay.
Austin.	Standard.	Mayfair.	Maudslay.
Beaufort.	Stuart.	Metallurgique.	28 Mors.
Belleville.	Sunbeam.	Minerva.	Seymour-Turner.
Belsize.	Swift.	Panhard.	Siddeley.
Bentall.	Talbot.	Robinson & Hole.	Vauxhall.
Benz.	Thornycroft.	Star.	Vinot.
Berliet.	West.		
Brown.			
Cadillac.			
C.G.V.			
Chenard & Walcker.			
Clément.			
Courier.			
Crossley.			
C.S.B.			
7 Cupelle.			
Darracq.			
Deasy.			
De Dietrich.			
De Dion.			
De la Buire.			
Delahaye.			
Dennis.			
Enfield.		*Parts driven with Chain.*	
F.I.A.T.			
Florentia.			
Ford.			
Gladiator.		Airex.	
Gobron-Brillié.		Beaufort.	
Gregoire.		Laurin & Klement.	
Humber.		10 Leader.	
Iris.		Napier.	
Isotta-Fraschini.		Panhard.	
Itala.		Star.	
30 James & Browne.		Sunbeam.	
Junior.		Vauxhall.	
Leader.			
Marlboro'.			
Martini.			
Maxwell.			
Mercedes.			
Mieusset.			
Morgan.			
15 Mors.			
Nagant.			
Napier.			
Nordenfeldt.			
Pilain.			
Radia.			
Rapid.			
Renault.			

IGNITION.				
L.T.M.	*H.T.M.*	*L.T.M. and H.T.*	*H.T.M. and H.T.*	*H.T.*
12 Albion.	Ariel.	24 Albion.	Argyll.	Adams.
12 Arrol-Johnston.	Aster.	Armstrong.	Bell.	Airex.
Belleville.	15 Austin.	24 Arrol-Johnston.	Belsize.	Alldays.
Bentall.	Beaufort.	25 Austin.	Brown.	Brotherhood.
Berliet.	Brooke.	Benz.	C.G.V.	Cadillac.
Bianchi.	Chenard & Walcker.	20 Darracq.	Climax.	Daimler.
Britannia.	Clément.	Delahaye.	Cupelle.	15 Enfield.
Bronhot.	Courier.	28 Mors.	60 Darracq.	Ford.
Crossley.	Deasy.	Nagant.	De Dion.	Humber.
C.S.B.	De la Buire.	Radia.	Dennis.	James & Browne.
80 Darracq.	18 Florentia.		25 Enfield.	10 Leader.
200 ,,	Germain.		Iris.	Marlboro'.
De Dietrich.	Gladiator.		12 Leader.	Maxwell.
F.I.A.T.	Gobron-Brillié.		Lindsay.	Mayfair.
40 Florentia.	Gregoire.		Mascot.	Napier.
Itala.	Hotchkiss.		Mass.	Robinson & Hole.
Junior.	Isotta-Fraschini.		Mieusset.	Rolls-Royce.
Laurin & Klement.	Maudslay.		24 Minerva.	15 Siddeley.
Martini.	Metallurgique.		Morgan.	10 Star.
Mercedes.	40 Minerva.		Siddeley.	Stuart.
Pilain.	15 Mors.		Spyker.	Vauxhall.
Thornycroft.	Nordenfeldt.		Sunbeam.	Vulpes.
Zust.	Panhard.		Talbot.	Winton.
	Rapid.		West.	
	Renault.			
	Rochet-Schneider.			
	S.C.A.R.			
	Seymour-Turner.			
	Simms.			
	Speedwell.			
	Standard.			
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